THE PRINCIPLES OF THE APPLICATION OF POWER TO ROAD TRANSPORT

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THE PRINCIPLES OF THE APPLICATION OF POWER TO ROAD TRANSPORT

(A SERIES OF SIX LECTURES DELIVERED AT THE TECHNICAL COLLEGE, FINSBURY)

BY

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PREFACE

At the invitation of the Finsbury Technical College the writer delivered, early in 1918, a series of six lectures on the *Principles of the Application of Power to Road Transport.* The present book represents those lectures in a somewhat expanded form.

Little has hitherto been written on this subject. Added to this, there is a paucity of published experimental data to serve as a substantial basis for design. Nevertheless, the writer has endeavoured to formulate a working theory based upon such tests as he found available or was able to make independently with the accelerometer. The work of the Technical Committee of the Royal Automobile Club has been found of much assistance.

Acknowledgment is gladly made of the courtesy of the Editor of *The Engineer* in permitting certain tables, etc., first contributed to his columns, to appear in this book. The reading of the proofs and the checking of the calculations has been kindly undertaken by Mr. H. G. Tisdall, B.Sc., A.M.I.C.E.

H. E. WIMPERIS

Hampstead
26th February 1913

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LIST OF SYMBOLS USED

A = Acceleration in feet per second per second, or area in sq. ins. or sq. ft., as the case may be.
B = Boiler pressure (gauge) in steam vehicles.
B.H.P. = Brake Horse-Power.
C = Cylinder capacity in cubic inches or in ccs.
D = Diameter of road wheels.
d = Cylinder bore in inches.
G = Gear ratio.
g = Gravitational constant.
I = Moment of inertia.
I.H.P. = Indicated Horse-Power.
l = Length of engine stroke in inches.
m = Ratio of rotational to translational momentum.
N = Revolutions per minute.
n = Number of cylinders in an engine.
P = Indicated mean pressure (except in Ch. I).
ηP = Brake mean pressure.
R = Tractive resistance in pounds per ton (except in Appendix I).
s = Distance in feet along hill slope for a vertical rise of 1 foot = $1 \div S$.
T = Torque in inch-pounds (except in Appendix I).
V = Velocity in miles per hour (except in Ch. I).
W = Weight in tons, or pounds, as the case may be.
γ = "Gear."
η = Mechanical efficiency.
ω = Angular velocity in radians per second.

USEFUL CONSTANTS

LENGTH, AREA AND VOLUME.

1 centimetre = 0·3937 inch.
1 inch = 2·540 cm.
1 sq. cm. = 0·1550 sq. in.
1 sq. in. = 6·452 sq. cm.
1 cu. metre = 35·31 cu. ft.
1 Imperial gallon = 4·546 litres = 10 lb. of water.
1 U.S.A. gallon = 3·785 litres.

WEIGHT AND PRESSURE.

1 kg = 2·205 lb.
g = 32·2 ft. per sec. per sec. = 981 cm. per sec. per sec.
1 atmosphere = 14·7 lb. per sq. in. = 760 mm. of mercury = 2,116 lb. per sq. ft. = 34 ft. of water.
1 lb. per sq. in. = 0·07031 kg. per sq. cm.
1 litre of water = 1 kg. = 1,000 grams.
1 metric ton = 2,205 lb.

ENERGY.

1 ft. lb. = 0·1383 kg. metre = $1{\cdot}356 \times 10^7$ ergs.
1 Joule = 10^7 ergs = 0·7373 ft. lb.
1 H.P.-hour = 1,980,000 ft. lb.
1 C.H.U. = 1,400 ft. lb.
1 B.Th.U. = 778 ft. lb.
1 Calorie = 3,087 ft. lb. = 2·205 C.H.U.

POWER.

1 watt = 1 volt. × 1 ampere = 10^7 ergs. per sec. = 1 Joule per sec.
1 kW = 1·34 H.P. = 0·239 calories per sec.
1 HP. = 0·746 kW = 76·04 kg. m. per sec.
1 metric H.P. = 0·986 English H.P. = 75 kg. m. per. sec.

OTHER CONSTANTS.

1 cu. ft. of water = 62·3 lb.
1 cu. ft. of air (N.T.P.) = 0·0807 lb.
1 radian = 57·3 deg.
$\log_e x = 2{\cdot}3026 \times \log_{10} x$.
$e = 2{\cdot}7183$.
Absolute zero = − 273° C. = − 459° F.
Atomic weights: O, 16; H, 1; C, 12.

THE PRINCIPLES OF THE APPLICATION OF POWER TO ROAD TRANSPORT

CHAPTER I

General survey of subject—Transport on land, sea and in air—Early construction of roads—Use of steam power—Internal combustion engines—Electricity—Resistance to motion in relation to loads, speeds, wheel diameters, nature of tires—Amount of power necessary—Watt-hours per ton-mile

MACAULAY in a well-known passage remarks :—

"The chief cause which (prior to the eighteenth century) made the fusion of the different elements of society so imperfect was the extreme difficulty which our ancestors found in passing from place to place. Of all inventions, the alphabet and the printing press alone excepted, those inventions which abridge distance have done most for the civilization of our species. Every improvement of the means of locomotion benefits mankind morally and intellectually as well as materially, and not only facilitates the interchange of the various productions of nature and art, but tends to remove national and provincial antipathies, and to bind together all the branches of the great human family. In the seventeenth century the inhabitants of London were, for almost every practical purpose, further from Reading than they now are from Edinburgh, and further from Edinburgh than they now are from Vienna."[1]

Since this passage was written, facilities for transport on land or sea have increased enormously. So far as the land is concerned the railway has been the almost universal means of carriage, but the great improvements in the main roads and in the mechanically-propelled

[1] "History of England," Vol. I.

vehicles travelling upon them has raised a challenge to this supremacy for, at any rate, certain classes of work. If we start in the most general way, we may say that there are three modes of transport possible—(1) on land, (2) on water, and (3) in the air. To haul a ton of goods on the surface of the land requires far more effort than on water. The most economical in power of all systems of conveying loads is carriage by sea, or by canal, at low speeds. It requires at least ten times the effort on land at similar speeds (even when railways are used), and far more at the usual speeds of motion. Water transport is not always available, however, and even when it exists, its use is sometimes prevented by the contrary interests of the railways. Air transport is the newest alternative, and is yet in the experimental stage. The effort needed for the conveyance of a ton of goods by this means is extraordinarily great—no less in fact, than if the load were hauled along the surface of the earth without even the intervention of wheels. This indicates how little prospect there is of the economical carriage of goods by this system in its present condition. Discovery will perchance bring to light some principle which may transform our notions on this subject, but of this there is, at present, little indication. The following comparative table illustrates these differences:—

	Effort in pounds per ton needed for propulsion.
Water transport at low speeds .	1
Rail transport.	15
Road transport	70
Air transport	300

The reason why road transport is able to compete with the railway, despite the above figures, is that the heavy capital costs of railway construction sometimes compensate wholly or partly for the extra outlay in power

necessary for road traction. Moreover, there are many cases where the quantity of goods to be carried is too slight to warrant a large capital outlay, particularly where the service is at all experimental. If a road transport system should prove unremunerative, it is easy to shift the fleet of vehicles elsewhere, but, once constructed, it is not possible to move a railway line. In the event of the service proving remunerative, it can be increased in capacity until, in the limit, it may prove to be of such size and permanence as to warrant the construction of a railway, when the road vehicles can be used elsewhere. Of course, this pre-supposes roadways to be in existence, or else to be capable of being easily and cheaply made. In quite new countries the railways are often built before the roadways, but in old settled countries there are always roads of some sort, and if the country be so fortunate as once to have been ruled by the Romans it will have at least the remains of a number of splendid highways—Roman roads—available for traffic. England was dowered with such a system of roads, but unhappily allowed them to sink into disrepair, until in the middle-ages the general method of conveying goods was by the poor means of pack animals along mere tracks. The lack of bridges was a very serious obstacle to wheeled traffic. Horse traffic was less impeded, since fords were numerous and at the worst it might be possible to swim a horse when wheeled transport would be impossible. But numerous delays arose when rivers were in flood and dangerous to cross, so that a wait of some days or even a week or more had sometimes to be faced. It is for this reason that on each side of the important fords villages were formed—thus accounting for the frequency with which we find old villages with their ancient churches at each end of a bridge, the modern bridge replacing the ancient ford. Even so late as the eighteenth

century the roads were allowed to remain in an inconceivably bad state. Memoirs of the time abound with references to this. Thus in a book written by Mr. C. F. T. Young in 1860, entitled "The Economy of Steam Power on Common Roads," mention is made that :—

"So late as 1763 there was but one stage coach from Edinboro' to London, and it set out only once a month, taking from twelve to fourteen days to perform the journey !—a distance now accomplished daily in the same number of hours."

This shows that in 1763 the Edinburgh coaches could only accomplish some thirty miles per day, which suggests terribly heavy roads, even if the daily running time were no more than five hours. The same writer also records that :—

"Until the close of the last (eighteenth) century, the internal transport of goods in England was chiefly performed by wagons, and was not only fearfully slow, but so very expensive as to exclude every object except manufactured articles, and such as, being of light weight and small bulk in proportion to their value, could bear the costs of a high rate of transport. Thus, for instance, the charge for carriage by wagon from London to Leeds was at the rate of £13 a ton, being 1*s.* 1½*d.* per ton per mile. Between Liverpool and Manchester it was 40*s.* a ton, or 1*s.* 3*d.* per ton per mile."

When we read of these obstacles to road transport it is not difficult to realise why waterways were favoured as an alternative. Pepys' Diary records the customary practice of getting to the London riverside houses by boat rather than by road ; and the City of Venice affords a striking illustration of the greater facility offered to the traffic of the middle ages by waterways than by roadways. Between that city and the mainland lies a swamp, and this obstacle offered more obstruction to her vast trade than the canals by which her produce could reach the sea and so find its way to distant ports. A striking political consequence was that Venice became part of the

Eastern Empire under Constantinople, in spite of the far greater proximity in mere distance of the city of Rome.

In England the first turnpike road was authorised in the reign of Charles II. It ran through Herts, Cambridgeshire and Hunts, and its upkeep was contributed to by those who used it. In Macaulay's History much is told of the conditions of the roads at the close of the seventeenth century. The following passages are so interesting as to be worth quoting :—

"It was by the highways that both travellers and goods generally passed from place to place. And those highways appear to have been far worse than might have been expected from the degree of wealth and civilization to which the nation had even then attained. On the best lines of communication the ruts were deep, the descents precipitous, and the way often such that it was hardly possible to distinguish, in the dusk, from the uninclosed heath and fen which lay on both sides. Pepys and his wife, travelling in their own coach, lost their way between Newbury and Reading. In the course of the same tour they lost their way near Salisbury, and were in danger of having to pass the night on the plain. It was only in fine weather that the whole breadth of the road was available for wheeled vehicles. Often the mud lay deep on the right and the left; and only a narrow track of firm ground rose above the quagmire. . . . The great route through Wales to Holyhead was in such a state that in 1685, a viceroy, going to Ireland, was five hours in travelling fourteen miles, from Saint Asaph to Conway. Between Conway and Beaumaris he was forced to walk a great part of the way; and his lady was carried in a litter. His coach was with great difficulty, and by the help of many hands, brought after him entire. In general, carriages were taken to pieces at Conway, and bourne, on the shoulders of stout Welsh peasants, to the Menai Straits. In some parts of Kent and Sussex none but the strongest horses could, in winter, get through the bog, in which, at every step, they sank deep. The markets were often inaccessible during several months. It is said that the fruits of the earth were sometimes suffered to rot in one place, while in another place, distant only a few miles, the supply fell far short of the demand. The wheeled carriages were, in this district, generally pulled by oxen. When Prince George of Denmark visited the stately mansion of Petworth in wet weather, he was six hours in

going nine miles; and it was necessary that a body of sturdy hindes should be on each side of his coach, in order to prop it. . . . The chief cause of the badness of the roads seems to have been the defective state of the law. Every parish was bound to repair the highways which passed through it. The peasantry were forced to give their gratuitous labour six days in the year. If this was not sufficient, hired labour was employed, and the expense was met by a parochial rate. That a route connecting two great towns, which have a large and thriving trade with each other, should be maintained at the cost of the rural population scattered between them is obviously unjust; and this injustice was peculiarly glaring in the case of the great North road, which traversed very poor and thinly inhabited districts, and joined very rich and populous districts. . . . On byroads, and generally throughout the country north of York and west of Exeter, goods were carried by long trains of pack-horses. These strong and patient beasts, the breed of which is now extinct, were attended by a class of men who seem to have borne much resemblance to the Spanish muleteers. A traveller of humble condition often found it convenient to perform a journey mounted on a packsaddle between two baskets, under the care of these hardy guides. The expense of this mode of conveyance was small. But the caravan moved at a foot's pace; and in winter the cold was often insupportable. . . . The rich commonly travelled in their own carriages, with at least four horses. Cotton, the facetious poet, attempted to go from London to the Peak with a single pair, but found at Saint Albans that the journey would be insupportably tedious, and altered his plan. A coach and six is in our time never seen, except as part of some pageant. The frequent mention therefore of such equipages in old books is likely to mislead us. We attribute to magnificence what was really the effect of a very disagreeable necessity. People in the time of Charles the Second, travelled with six horses, because with a smaller number there was great danger of sticking fast in the mire."[1]

IMPROVEMENTS IN ROAD CONSTRUCTION.—The first to remedy the condition of English roads was the well-known engineer Macadam. Macadam during the earlier part of his life had been interested in road making, but it was not until nearly sixty years of age that he gave

[1] History of England, Vol. I.

himself up to it entirely, filling the position of surveyor-general of the Bristol roads. His memory is preserved in the term "macadamised road," now about to be superseded by the modern dustless type of road. Sir John Macdonald, the modern exponent of road construction, records that :—

"It was only within the last century that science came in for the first time, when Macadam and Telford worked out a very efficient system of road construction, but having built the foundation of their road and put down their stones, they left it to the road user to do the rolling, and in order to prevent the formation of ruts they put up trestles which forced the user to go in a zigzag fashion. When the road had been well rolled the trestles were transferred to the opposite side. The original roads of Macadam and Telford were good ones, but they came to be neglected when the railway was invented. Unfortunately present-day methods of road rolling had nullified the intentions of Macadam, which were that the road should be constructed absolutely water-tight. Macadam had laid it down that nothing should be used as a binding material for the stones which would absorb water, but the neglect of this was the cause of the formation of one of the most serious objections to the present so-called macadamised road; viz., pot-holes."[1]

Modern roads have to be constructed in an entirely new way to suit modern traffic conditions. Motor traffic has grown so greatly in volume as to make the suitability of the road to horse traffic of secondary and no longer primary importance. Nevertheless, there is no reason why roads should not be made suitable for both. The disadvantage of horse traffic from the road point of view is the disintegrating effect of the rapid blows from iron-shod hoofs and the very high pressure between the iron tyre and the road. With motor traffic these are absent, but there is an additional factor to be considered in the obliqueness of the reaction between the driving wheels and the road. These wheels are trying to push the road away behind

[1] *B. A.*, 1912.

them. All these problems have been very carefully considered by the Road Board, and by their engineer, Col. Crompton. We have already referred to what Sir John Macdonald, who is a member of the Road Board, has written of the roads built by Macadam, and it is not out of place to give his description of what are considered to be desirable features in modern road construction :—

"The essential of a good road was that no water should ever enter under the surface. Macadam had laid it down that the stones used for road purposes should not weigh more than six ounces, but many now used weighed several pounds. The problem before the Road Board was to find a surface which would not break, a surface which would be easier for the carrying of passengers and goods, and one which would not form pot-holes nor exude mud. With this end in view, the Road Board had been making laboratory experiments by means of special apparatus at the National Physical Laboratory. A popular idea seemed to be that by ordering a few barrels of tar, digging up the road, and then pouring on the tar and rolling it would give a good road. This was quite wrong. They had not only to provide a good foundation for the top surface, but it was also necessary to have a top surface which would prevent noise as far as possible. The Road Board had come to the conclusion that a road should be capable of construction so that when properly made it would only need renewing so far as the top dressing was concerned.

"Experiments now being carried out were gradually evolving the proper foundations and top dressings for different classes of roads according to the volume of traffic over them. Such roads would be thoroughly water-tight. The initial outlay was considerably higher than that of the mud roads now existing, but the lasting qualities were infinitely greater." [1]

EARLY VEHICLES—The production of fairly good roads, coupled with the invention of the steam engine, led to many efforts being made in the early part of the nineteenth century to use steam power for road transport. They all came to nothing, however, largely on account of the great weight of the mechanism and of its unattractive-

[1] *B. A.*, 1912.

ness to those who lived on or near the routes traversed. This lack of attractiveness will be understood by those who have had experience of the old steam tramcars formerly seen in some provincial cities. The horse transport interests also offered a strenuous opposition to the new system.

For the transport of heavy loads, however, steam was eventually found to be quite suitable, and gradually there was evolved the modern steam traction engine and steam lorry. An interesting sidelight is thrown on the difficulties met by those who were endeavouring to develop steam carriages in the "thirties" of last century by the following details which Mr. C. F. Dendy Marshall has collected from old volumes of *The Mechanics Magazine*. There is a striking lack of co-operation—to put it no more strongly—between the road authorities and the steam carriage inventors.

The extracts are as follows :—

"*April 5th*, 1834

"RUSSELL'S STEAM CARRIAGE—A new steam carriage (Mr. Russell's) commenced plying between Glasgow and Paisley on Wednesday. The carriage is attended by a supplementary vehicle, containing the necessary supply of charcoal and water. The carriage itself is superbly fitted up, holds six inside and twenty outside passengers, and is hung on springs, quite free of the boiler and machinery.

"ROBERTS' LOCOMOTIVE CARRIAGE—Mr. Roberts, of the firm of Sharp, Roberts and Co., engineers, of Manchester, has been for some time engaged in the construction of a locomotive carriage for common roads, for which he has obtained a patent. An experimental trip is described, in the course of which the vehicle attained a speed of twenty miles an hour, with upwards of forty passengers, the 'acclivities of the road' being mounted without any effect on the speed. A few days later its career was brought to an abrupt conclusion by the bursting of the boiler, which blew in the front of a shop and injured one or two people.

"*May 3rd*, 1834

"The service between Glasgow and Paisley is described as 'fully and permanently established,' the carriages running regularly at ten miles an hour. They appear, however, to have stopped running by the 16th of May.

"*July 19th*, 1834

"The Glasgow and Paisley carriages had by this time made another start, sometimes attaining a speed of seventeen miles per hour. They were very popular, being always crowded with passengers. The road trustees immediately laid down new metal, the thickness at first being insufficient to stop the coaches. They then employed horses and carts and a number of men during the night, depositing stones to such a depth that they were obliged to cut away the bottom of the toll-gate to allow it to close over the mass.

"*July 26th*, 1834

"The hostile proceedings of the road trustees last mentioned seem still to have been unsuccessful, as the coaches ran regularly three trips each day, sometimes carrying thirty-nine passengers.

"*August 16th*, 1834

"One of the Glasgow and Paisley carriages broke a wheel, overturned and burst its boiler, killing five persons, in consequence of which the service was interdicted by the Court of Session."[1]

In 1895 Daimler brought out his high speed engine for automobiles. This was an internal combustion engine, using petrol as its fuel and working on the four-stroke cycle. In the eighteen years that have elapsed since that important discovery immense progress has been made in the development of the motor-car and motor wagon, of which there are now more than a million in use in England and the United States. Many fuels besides petrol have been used in these vehicles, such as paraffin, benzol and alcohol, but petrol is the best and would be used universally were it sufficiently cheap. Electricity has also been brought into service, either as a source of power or as a means of utilising to the best advantage the power provided by the engine on the vehicle. The latter

[1] *R. A. C. Journal*, January 3rd, 1913.

vehicles are commonly known as petrol-electric, although there is no reason why they should be particularly associated with engines using petrol to the exclusion of other fuels. The former vehicle—the purely electric vehicle—carries a battery of accumulators and is propelled by one or more electric motors worked by the current. Electric storage vehicles are not much used in this country, though their much greater success in America suggests that they may some day become more popular here. Recently a number of successful efforts have been made to run purely electric vehicles (trolley omnibuses) on the road, which draw their current from overhead conductors. In this case the power is derived from a central generating station.

CONTACT OF TIRE AND ROAD—With this preliminary on the construction of roads and the general nature of the vehicles running on them, we proceed to the discussion of the nature of the action between tire and road. Here our authority is Mr. H. R. A. Mallock, F.R.S.[1] When a loaded wheel rests or rolls on the road the average pressure between the two is equal to the total load divided by the area of contact. In the case of pneumatic tires this pressure will be nearly uniform over the whole area, but with iron tires or solid rubber tires the pressure will be some 50 per cent. higher in the centre and fall off towards the edges of the contact area. If this pressure should at any point in the road exceed the elastic limit a certain depth of road surface is destroyed every time this pressure occurs. It is therefore important to limit the amount of this pressure. Mr. Mallock finds that the angle of contact between tire and road is

$$= \frac{c}{\sqrt[3]{r^2.b}}$$

[1] See *Proc. Inst C.E.*, Vol. 178.

where c = a constant, r = radius of wheel and b = breadth of wheel. The area in contact will therefore be

$$= c\sqrt[3]{r.b^2}.$$

As the result of experiment he finds the following figures for the pressure :—

TABLE I—LENGTH OF CONTACT AND INTENSITY OF PRESSURE BETWEEN VARIOUS PAVING MATERIALS AND A STEEL TIRE 1 INCH WIDE CARRYING A LOAD OF 1 TON

Radius of Tire.	Steel Surface.			Granite.			Jarrah Wood.			Deal.			Asphalt.		
	L	p	P	L	p	P	L	p	P	L	p	P	L	p	P
Ins.	In.	Lb.	Lb.	In.	Lb.	Lb.	In.	Lb.	Lb.	In.	Lb.	Lb.	In.	Lb.	Lb.
24	0·68	3300	4950	0·74	3000	4500	0·81	2780	4170	0·95	2350	3520	1·18	1900	2350
18	0·625	3600	5400	0·69	3250	4900	0·74	3020	4550	0·87	2560	3820	1·08	2080	3120
12	0·54	4150	6200	0·59	3800	5700	0·64	3500	5250	0·75	3000	4500	0·93	2480	3720
6	0·41	5400	8100	0·45	5000	7500	0·49	4500	6750	0·58	3850	5750	0·74	3000	4500

Table I shows the order of the arc of contact, and the intensity of pressure to be expected when wheels of various radii, one inch wide, carry a load of one ton on various kinds of material used in paving, where

L denotes the length of the arc of contact.

p ,, mean pressure per square inch.

P ,, maximum pressure per square inch.

The experiments from which the results in Table I are derived were made by placing hard-steel arcs of twenty-four inches and six inches radius, respectively, on flat surfaces of the various materials tested, with a thin strip of carbon tissue paper and copying-paper interposed, the load being then applied in a hydraulic press. The blackened copying paper showed the area over which contact had prevailed ; and since the steel used was much harder than any of the materials tested, its curvature remained practically unaltered.

Some measurements, given in Table II, were made by Mr. Mallock on solid tires, using carbon tissue paper as the means of record, and show that the contact-area of each tire of the motor-omnibuses running in London is about thirteen square inches, giving a mean pressure of 120 lb. to 140 lb. per square inch.

TABLE II—AREA OF CONTACT BETWEEN THE GROUND AND THE SOLID INDIA-RUBBER TIRES OF A MOTOR-OMNIBUS

The area of contact A is an oval, *a* denoting the long axis, and *b* the short axis, of the oval.

Load on each front wheel = 0·84 ton nearly.

„ „ „ back wheel = 1·46 „ „

Front Wheel.

	a	*b*	A	*p*	P
	Ins.	Ins.	Sq. Ins.	Lb.	Lb.
New tire .	6·9	2·5	13·5	140	230
Worn tire .	5·7	3·4	14·6	129	200

Back Wheel.

	Outer Tire.					Inner Tire.				
	a	*b*	A	*p*	P	*a*	*b*	A	*p*	P
	Ins.	Ins.	Sq.Ins.	Lb.	Lb.	Ins.	Ins.	Sq.Ins.	Lb.	Lb.
New tire .	7·6	2·2	12·6	129	200	7·65	2·1	12·6	129	200
Worn tire .	5·7	3·16	14·2	114	180	5·5	3·35	14·4	113	179

With pneumatic tires it is nearly correct to take the pressure on the ground as uniform over the whole area of contact, and equal to the air-pressure in the inner tube. Thus the area of contact is obtained by dividing the load by the internal pressure.

Of course the smaller the air-pressure in the tire the less intense is the pressure on the road. A low air-pressure therefore is good for the roads, but on the other hand, it is bad for the tires, because of the increased distortion of the tire-fabric. The intensity of the pressure with pneumatic tires is in any case so low that the ratio of air-pressure to load, which Mr. Mallock considers will

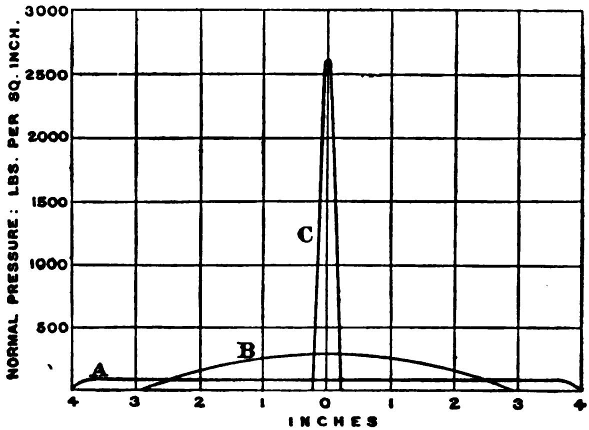

FIG. 1.—Diagram due to Mr. Mallock, showing intensity of pressures between road and tire when (A) pneumatic, (B) solid rubber, and (C) steel tires are fitted.

at some future time be taken as the standard, will be settled rather with reference to the tire than to the road. In Fig. 1 a comparison is given of the intensity of the pressures on the road due to the same wheel fitted with iron, solid india-rubber, and pneumatic tires, under the same load.

RESISTANCE TO MOTION—The resistances to motion on a level road are :—

(1) Rolling resistance.
(2) Internal resistance of the mechanism.
(8) Air resistance.

The total of these three may very suitably be called the tractive resistance, since it represents the amount of tractive force which must be exerted by the engine.

Gravitational resistance only comes in when the road is not level, and it is, of course, the same for all vehicles of the same weight when on the same road. It depends solely on the slope, so that a gradient of 1 in 10 produces a gravitational resistance of one-tenth of a ton per ton, or 224 lb. per ton.[1] And to find the total resistance the tractive and gravitational resistances have to be added together.

The rolling resistance is made up of two parts, one due to the road surface and one to the tire. Part of the energy wasted goes into grinding or churning up the road and part into heating the tire. The former part is usually assumed to be independent of the speed of motion, though there is reason to suspect that when the roads are very wet it may be greater at high speeds than at low ones. The resistance due to the rolling of rubber tires was shown by the experiments of Ewing and of Bourlet, many years ago, to be independent of the speed. But the hysteresis loss in the rubber may be expected to rise with increase of load. There is also a similar hysteresis loss in the road itself.

Mechanism resistance depends entirely on the amount of mechanism there is attached to the road wheels. Thus a four-wheeled hand-cart having its wheels entirely free from all mechanism but their bearings would encounter practically none of this resistance at all. Whereas, in a steam road-roller which has its rear road wheels coupled up to the fly wheel and engine, the effort needed to haul it would always be much larger than if the engine were disconnected from the wheels. These are the extreme cases of freedom and lack of freedom. The ordinary motor car comes between the two. The motor car

[1] NOTE.—Here as elsewhere in the book, the ton of 2240 lb. is taken.

engine can be engaged, through the clutch, with the road wheels when desired, and disconnected when it is desired to coast; though even when disconnected in this way the rotation of the rear road wheels also requires the rotation of the back axle, the bevel or worm, and the propeller shaft together with certain of the gear box mechanism. If the effort necessary to haul a motor car were being measured it would be necessary for this reason to bear in mind that the requisite effort in pounds per ton would be not the effort needed at the road wheels, but the effort needed at the clutch. The resistance of the mechanism is largely independent of the load, though somewhat dependent on speed, but the loss in the gear box due to churning of the lubricant increases rapidly with the speed—in fact, the resistance due to this churning is almost proportional to the speed squared, and the energy loss to the speed cubed.

The air resistance, unimportant at low speeds, is much the most important at high speeds. At the very high speeds attained by racing cars, almost the whole resistance is due to the air. If a flat square area be exposed to a wind blowing straight at it the pressure on the area will be given by the equation

$$F = 0{\cdot}003V^2A$$

where F is the force in pounds, V the velocity in m.p.h., and A the area in square feet. Now a motor vehicle, although it presents some area to the wind, certainly does not present a single flat square area. Its line is very much broken, and there are usually several curiously-shaped areas one behind the other. So that the total force due to air resistance is sometimes far greater than that given by putting A as merely equal to the "projected" area of the vehicle. But since body shapes vary so much, it is best to make actual measurements whenever it is possible to do so, and not to rely on any rule.

The sum of the components gives the tractive resistance, commonly stated in pounds per ton. This form of statement may seem to imply that all three components rise and fall with the load, which is not true of the air resistance, since in a closed-in motor wagon it would be the same whether its interior were occupied by a load or not. So that, although it is convenient to measure tractive resistance in pounds per ton, it must be remembered that the air resistance will depend not on the weight of the load but the shape of the body used to carry it. The effect of wheel diameter upon the tractive effort is not large. Increase of wheel diameter decreases the road resistance component, since the pressure between road and wheel is then less, and the centre of gravity of the vehicle keeps a more horizontal path, and there is less hammering on the road. Hammering of this kind is more productive of loss in the wheels when rubber tires are used than steel ones, since the hysteresis loss in rubber is greater. Nevertheless, the comfort of travelling is so much increased by the use of rubber, and the life of the engine is so much prolonged (by being protected from vibration), that for all loads up to four or five tons the use of rubber tires (pneumatic or solid) is almost universal.

The tractive resistance R, expressed in pounds per ton, is usually calculated from an equation of the following type :—

$$R = a + bV + cV^2$$

where a is the resistance due to the components independent of V; b is the item added by road hammering—but it is usually so small compared with the others that this term is omitted; and cV^2 is the component due chiefly to the air resistance but increased also by lubricant churning in the gear box, and probably by mud churning by the wheels when the vehicle is running on very heavy

roads. If W be the total running weight of the vehicle, we may therefore write

$$RW = aW + cWV^2 = \text{total force opposing motion.}$$

AMOUNT OF POWER NECESSARY—The amount of power necessary for the propulsion of vehicles is usually measured in horse-power, or in the case of electric vehicles in kilowatts.

When a machine is capable of doing 33,000 ft. lb. of work every minute (or 550 ft. lb. every second), it is said to be of 1 H.P. A machine capable of doing 1,650,000 ft. lb. of work per minute would be a machine of 1,650,000 ÷ 33,000 = 50 H.P. If, therefore, a vehicle weighing, say, 4 tons be moving along a level road at a speed of 15 ft. per second, and if the tractive resistance be 50 lb. per ton, the H.P. is calculated thus :—

$$\text{H.P.} = \frac{4 \times 50 \times 15 \times 60}{33,000} = 5\tfrac{1}{2} \text{ H.P. approx.}$$

Or more generally, if the total moving weight be W tons ; the tractive resistance R lb. per ton ; and the speed in miles per hour V, then

$$\text{H.P.} = \frac{\text{WRV} \times 88}{33,000} = \frac{\text{WRV}}{375}.$$

A racing car weighing a ton may attain a speed of 100 m.p.h. when R is perhaps as high as 350 lb. per ton, and in that case

$$\text{H.P.} = \frac{100 \times 350}{375} = 94.$$

If such a car had to climb a gradient of 1 in 15 its speed would be reduced to perhaps 70 m.p.h., in which case air and road resistance might be 220 lb. per ton, so that the total resistance would be

$$\left(220 + \frac{2,240}{15}\right) = 220 + 150 = 370$$

and $$\text{H.P.} = \frac{70 \times 370}{375} = 69.$$

ELECTRIC POWER—The unit of electric power is the kilowatt, usually written kW., which is equal to 1,000 watts. A watt is the power given by a current of one ampere, which flows between points at a difference of pressure of one volt. If, therefore, I be the intensity of the current in amperes and V the pressure difference in volts, the

$$\text{kW.} = \frac{V \times I}{1{,}000}.$$

Thus, if a current of 100 amperes under a pressure of 500 volts be supplied to an electric motor, the kW. supplied will be

$$\frac{100 \times 500}{1{,}000} = 50 \text{ kW.}$$

The kW. unit of power is larger than the H.P. and

$$0{\cdot}746 \text{ kW.} = 1 \text{ H.P.}$$

in other words, 746 watts go to 1 H.P. We spoke above of 50 kW. being supplied to an electric motor. Such a motor would have an efficiency of somewhat over 75 per cent., but for motors within that size it is roughly true to say that an electric motor will yield about as many H.P. in mechanical work as it receives in kW. of electric power.

The product of power by time gives, of course, the work done. Thus 1 H.P.-hour = 33,000 × 60 = 1,980,000 ft. lb. Similarly the work done in one hour by a power of 1 kW. is 1 kW.-hour; or by a watt for an hour is 1 watt-hour. These are often convenient. It is usual, for instance, to speak of the watt-hours of energy given to an electric vehicle; and sometimes this is divided by the ton miles, *i.e.*, by the product of tons weight by miles travelled. In that case we come across the expression "watt-hours per ton mile." Now this is a very important kind of measurement. Watt-hours we know to be energy. This we now divide by weight and by distance.

Dividing energy by distance gives us force. So that "watt-hours per ton mile" is really a force per ton of weight, *i.e.*, it is of the same dimensions as pounds per ton of tractive resistance. Now it happens that

2 watt-hours per ton mile = 1 lb. per ton

very nearly. And this is an important relationship which we will proceed to establish.

$$2 \text{ watt hours} = \frac{2}{746} \text{ H.P. hours}$$
$$= \frac{2}{746} \times 33{,}000 \times 60 \text{ ft. lb.}$$
$$= 5308 \text{ ft. lb.}$$

so that

2 watt-hours per ton mile = 5,308 ft. lb. per ton mile, but there are 5,280 ft. in a mile

$$\therefore 2 \text{ watt-hours per ton mile} = \frac{5{,}308}{5{,}280} \text{ lb. per ton}$$

or 2 watt-hours per ton mile = 1 lb per ton almost exactly.

If, therefore, we know the resistance to a given vehicle at a given speed to be 45 lb. per ton, we know that it will need 90 watt-hours per ton mile even if the electric motors were of 100 per cent. efficiency. This relationship affords a useful check upon extravagant statements.

CHAPTER II

Measurement of power — I.H.P., B.H.P.—Torque — Efficiency — Road Tests—Measurement of speed and resistance—Use of accelerometer—Loss of power in engine friction and in transmission gear—G.T.M.

MEASUREMENT OF POWER—It is important to be able to measure accurately the amount of power given by the different forms of engine used in road vehicles. There are two ways of doing so

(1) By bench tests.
(2) By road tests.

In the former the engine is mounted on the test bed in the works, and is loaded by some artificial load so that it exerts its full power, or such proportion of full power as may be desired. There are various ways of applying the load.

(1) By friction brake ;
(2) By electric generator ;

and the former may be concerned with solid friction, or fluid friction. The commonest method is that shown in Fig. 2, where a fly-wheel has a number of wooden brake-blocks loosely fitted to the rim and connected together by one or more ropes or by a canvas belt. A number of heavy weights are hung as at P_1 and a spring balance is placed at P_2. The distance D is measured in feet. As the force due to the weight P_1 is greater than the force at P_2, there will be a net force resisting the rotation of the flywheel of $(P_1 - P_2)$. The work done, therefore,

by the fly-wheel in one revolution in the direction of the arrow = force × distance moved

$$= (P_1 - P_2) \times \pi D$$

and if the r.p.m. be N, the

$$\text{B.H.P.} = \frac{(P_1 - P_2) \times \pi D \times N}{33{,}000}.$$

As this power is all being spent in friction, it produces heat, and in the larger machines the fly-wheel has to be cooled by a water spray.

Sometimes the solid friction of the brake blocks is

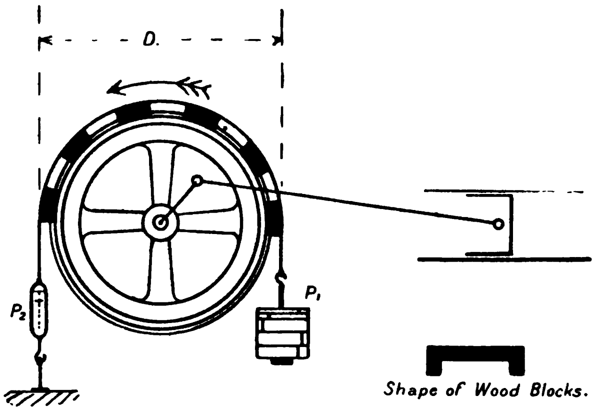

FIG. 2.—B.H.P. Tests. Hand-brake Method.

replaced by the fluid friction of a water brake, in which case a rotor with fixed vanes is made to churn up water in a cylindrical casing. The casing has adjustable vanes fastened to it and the action is very similar to that of an ordinary centrifugal pump. This reaction produces a tendency to rotate the casing, and this effort is balanced by a steelyard and weight—so that the turning effort is measured. If this torque be T lb. ft., then the

$$\text{B.H.P.} = \frac{T \times \text{angular velocity}}{550} = \frac{2\pi NT}{33{,}000}.$$

Air may be used in place of water and in that case no outside casing is required. The fan is so made that the size of the paddles can be altered, and for each size of paddle the B.H.P. needed to rotate it at various speeds is known from previous measurements. This is a convenient method so far as simplicity of apparatus is concerned, but it does not enable tests at low power to be made except by the use of a very wide range of blade sizes—and since the blades cannot be changed while the engine is in motion, the test is a lengthy one. With the brake drum method or the hydraulic brake, it is possible to vary the resisting torque at will. A further objection to the air brake is the fact that the calibration scale of B.H.P. will differ according to the degree of freedom which the air has to get away from the fan.

In the electric method—which is also the best method—the engine is made to rotate a dynamo and measurements are made of the volts and amperes of the current generated. Then we know that the output of power in kW. And the B.H.P. can be deduced by dividing the kW. by 0·746. An allowance has however to be made for the losses in the dynamo. No electric generator converts quite all the mechanical power it receives into electric power. The efficiency of conversion is always less than 100 per cent., in fact usually nearer 80 per cent., so that the efficiency of the generator must be known in advance, and an allowance be made accordingly in calculating the power generated by the engine. An ingenious method of simplifying this test is to mount the field magnets also on bearings and measure the torque tending to rotate them just as in the hydraulic brake method.

In one or other of these forms of bench test we can obtain the engine torque at various speeds and so plot what is known as a " torque curve." We shall return to

this shortly, and in the meantime discuss the usual ways of "indicating" an engine, *i.e.*, obtaining from it a diagram showing the relationship of the pressure on the piston with the position of the piston in its stroke. These diagrams enable us to see how the engine is working and to measure the amount of power given to the piston (I.H.P.). This can then be compared with the B.H.P. and we can see whether the engine is absorbing

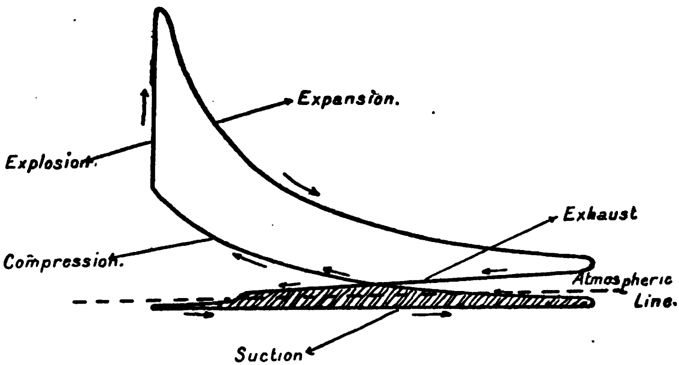

FIG. 3.—Indicator Card for Four-stroke-cycle Internal Combustion Engine.

too large a proportion of power in its own internal mechanism.

The typical indicator diagram from a steam engine and the types of indicator instrument used to obtain it, are so well known as not to need description here. The indicator card for an ordinary four-stroke internal combustion engine is shown in Fig. 3, in which the negative area is shown shaded. Owing to the high speed of these engines it is impossible to take diagrams by the older methods and special forms of indicator are needed. One such is shown in Fig. 4. This is known as a reflecting

indicator, of which an early type, due to Prof. Perry, was made and used in the Finsbury College. In the form shown in the diagram, which is due to Prof. Hopkinson, there is a small cylinder containing a piston which is fastened to a piston rod which bears against a small straight beam, held at the ends (in some instruments the spring is in the form of a diaphragm against which the gas pressure acts without the intervention of a piston). The pressure bends this spring upwards and so tilts the little mirror

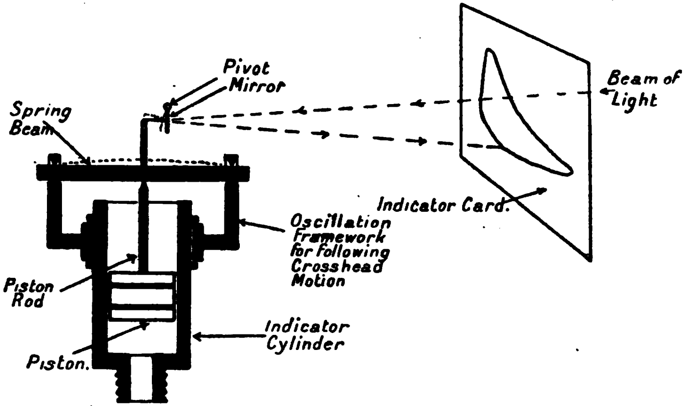

FIG. 4.—Reflecting Indicator. Hopkinson Type.

about a pivot. A beam of light is made to shine on the mirror, and the tilting of the latter deflects the path of the reflected beam through twice the angle through which the mirror is tilted. The reflected beam, therefore, moves through an angle directly proportional to the gas pressure. To give the beam a sideways motion equivalent to the stroke of the engine, the rocking lever is made to reciprocate the bracket carrying the spring beam and the mirror. The mirror, therefore, gets a partial rotation about a vertical axis proportional to the stroke. This being so, the beam of light on being reflected from

the mirror draws a true indicator diagram on the screen, and owing to the speed at which the spot of light moves, the diagram is seen as a continuous curve. For permanent record, a photographic plate replaces the screen.

In this way quite accurate indicator cards can be obtained even at the highest engine speeds. Not many works have this equipment, however, and the taking of such diagrams from high-speed petrol engines is not yet common practice. Nevertheless, it affords an admirable means of actually seeing the effect on the indicator diagram of varying the positions of the control levers.

The ratio of B.H.P. to I.H.P. is, of course, the mechanical efficiency of the engine, and their difference gives the amount of engine friction. It has been found by experiment that at any given speed the power lost in engine friction is almost independent of the load. Thus, in a special test of a gas engine of 76 I.H.P. which was kept very nearly at 200 r.p.m. all the time, the power lost in engine friction was

10·8 H.P. at no load.
11·0 H.P. at half load
11·7 H.P. at full load.

Numerous tests by Prof. Hopkinson and others also confirm this important result.

TORQUE CURVES—We now return to the B.H.P. test and the torque curve obtained from it. A typical petrol engine torque curve is shown in Fig. 5. Torque, or turning moment, is measured by multiplying the force by the length of the arm about which it acts. It is usual to plot the torque vertically and engine speed horizontally—the curve shown in the diagram fits the test of a 15-H.P. engine. It will be seen that the torque rises to a maximum at about 900 r.p.m. and then falls away; this marked falling off is seen in all but racing engines, and is due to the difficulty of the air and fuel getting through the valves

during the short time they are open. The effect is often accentuated by the valve lift being intentionally made small so that the engine cannot be driven at too high a speed. The reason why the curve is lower at 400 r.p.m. than at 600 r.p.m. is that at the slower speed the gases have more time to cool, and therefore to lose pressure.

It is a peculiar and little noticed fact that the shape of the left half of the curve must depend on the way in which the torque measurement is made. Assume, for

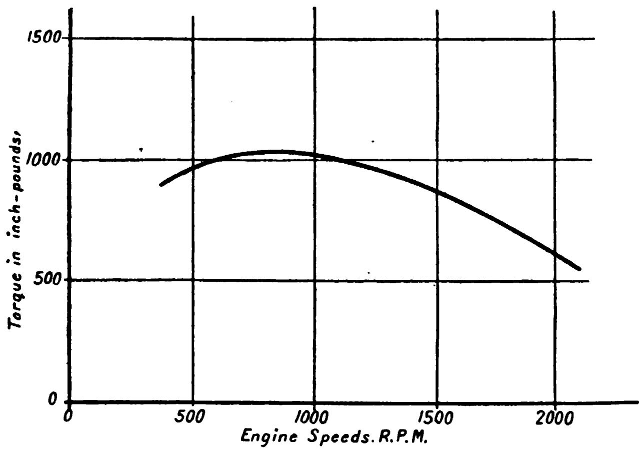

FIG. 5.—Typical Torque Curve.

instance, that this engine was being tested on the bench under a friction brake, and let the resisting torque be adjusted to 1,000 inch-lb. and assume also, as is probably the case, that this resistance would be the same at all speeds of working. Now look at the diagram in Fig. 5 and notice that the horizontal line corresponding to 1,000 inch-lb. cuts the torque curve at 600 and 1,100 r.p.m. At either of these speeds, therefore, the engine effort and the frictional resistance exactly balance. Will the engine, therefore, run at either of these speeds? At

first one expects that it would, but on looking closely into it one sees that it could not run at 600, and that any effort to make it do so would be unsuccessful. This is because the engine speed must vary slightly and at 600 r.p.m. the least tendency to decrease speed would mean that the engine torque would be less than the resistance and the speed would fall yet more, and the more it falls the more rapidly it does so, until it stops altogether. Also, if the speed rise slightly above 600, the engine effort becomes larger than the resistance and the speed tends to rise still more. The 600 r.p.m. is, in fact, a point of unstable equilibrium, and none of the left-half of the curve could be obtained by this method. Now consider the crossing point at 1,100 r.p.m. A momentary decrease of speed brings us to a point at which there is an excess of engine effort tending to restore the speed to its original figure. Now suppose the speed accidentally rises above 1,100. Then there is an excess of resisting effort and the speed falls back to 1,100 again. This, therefore, is a point of stable motion.

It follows that when the engine is set to work against a constant torque it can only run on the right half of the curve. If, however, the resisting torque were one which rose rapidly with the speed, say, from zero at zero r.p.m. to say 1,000 at 500 r.p.m., then there would only be one crossing point, viz., that at about 400 r.p.m., and this point will be seen to be one of stable motion. A resistance therefore, which rises rapidly with speed, as in a fan brake, an hydraulic brake, or an electric brake, makes it possible to get the left hand part of the torque curve. In the case of a motor vehicle the resistance is made up of a part which is independent of the speed, and a part which rises rapidly with speed, so that an intermediate state of affairs is arrived at and some part of the curve to the left is available.

ROAD TESTS.—Tests are also made with the complete vehicle on the road, and the information obtainable on the bench test is also obtainable on the road. Such tests are kept as closely as possible to normal working conditions. Measurements are needed of speed and H.P., also of acceleration, of hill-climbing ability, and of braking power.

To measure the B.H.P. on the road it is necessary to know the speed and the tractive resistance (R) in pounds per ton.

Then, see p. 18,

$$\text{B.H.P.} = \frac{\text{velocity in m.p.h.} \times \text{R} \times \text{weight of vehicle in tons.}}{375}$$

The speed is usually read on a speed indicator, and the resistance[1] by an accelerometer. The action of the speed indicator is well known. That of the accelerometer is to weigh the forces which oppose the motion of the vehicle; the needle of the instrument points to the figure on the scale, showing the number of "pounds per ton" of tractive resistance at any moment. Such devices are called accelerometers because first introduced for the measurement of train acceleration, but they have many other uses.

Until 1909 there were three chief forms of accelerometer —those of Desduits, Lanchester and Trotter—but in spite of their individual excellencies none of them were suitable for road transport work. In 1909 the author, with the skilful aid of Messrs. Elliott Bros., constructed an accelerometer on a new method altogether, which would easily be able to endure hard usage on the road, while losing nothing in accuracy. This instrument will be described.

[1] The older way of measuring tractive effort was by hauling the vehicle with a spring draw-bar attached to another vehicle. This was found to be a difficult measurement to make, particularly at high speeds, and the least hill slope affected the readings seriously.

ACCELEROMETER—The general appearance of the instrument is seen in Fig. 7, and a diagrammatic representation of its mechanism is given in Fig. 6. To make a measurement of the tractive effort necessary to overcome the resistance to motion the instrument is levelled on the floor of the car by means of the adjustable leg, and then, when the car is coasting, the indication of the needle gives the reading in " pounds per ton." This is the tractive resistance.

The general nature of the internal mechanism may be

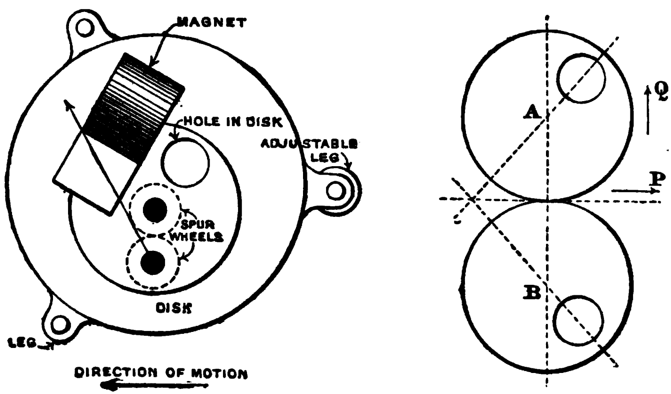

FIG. 6.—Mechanism of Accelerometer shown.

described as a lop-sided copper disc, mounted on a vertical axis and controlled in its rotation by a coiled spring. Any acceleration causes the heavier side of the disc to lag behind, and so partially wind up the spring. The degree to which the spring is wound up measures the acceleration. Any tendency of the disc to oscillate is checked by a magnetic field at right angles to the plane of the disc. Besides these parts there is what is called " the compensating balance," which causes the instrument to record absolutely correctly, even when travelling around railway curves, or when on a road heavily cam-

bered to one side or the other. In Fig. 6 the copper disc is shown with a hole cut in it near the circumference, which has the effect of throwing the centre of gravity slightly out of the centre of figure. On the pivot of the disc is fastened a spur wheel which gears in with another equal spur wheel mounted on a parallel axis and carrying the pointer. This pointer moves over the scale shown in Fig. 7. It will be seen that there is a small permanent magnet placed so as to "damp" the motions of the copper disc without having any of the "sticking" qualities which accompany frictional damping. The spring, which is coiled up by the rotation of the disc, is not shown, but it lies in the horizontal plane just above the disc. In later forms of the instrument the arrangement described above has been varied slightly, but not so as to affect in any way the general principle of working.

The forces acting at the centre of gravity of the disc are directly proportional to the impressed acceleration, but as the disc rotates through angles of 30 degrees, 45 degrees, 60 degrees, etc., the "arm" gets less, and the couple twisting the spring does not rise so rapidly as the acceleration producing it. The scale has to be graduated in accordance with this fact, and the divisions close up when the needle has to move through large angles. The law which governs this is that of the change of the ratio $\frac{\theta}{\cos \theta}$.

It is now necessary to describe the mode of action of the compensating device. If, whilst the disc is deflected by an acceleration, a second acceleration (or slope) should be acting at right angles to the first, there will be an additional couple tending to wind (or unwind) the spring. The needle must then give false indications of the acceleration which it is desired to measure. This would be a very serious fault, since such transverse accelerations are very common in practice and often of considerable

amount, sometimes as much as 10 feet per second per second. (This type of error is noticeable in the pendulum forms of instrument, where the complicating vertical acceleration often far exceeds that of gravity itself.) To get over this difficulty the two gear wheels, already mentioned, were added, and it was so arranged that the moments of mass about the two respective axes should be equal. The equality of mass moments makes the system equivalent to two equal copper discs geared together at their circumferences, and each having their centres of gravity eccentric to the same extent. The effect of this is illustrated in the diagram to the right-hand side in Fig. 6. Inspection of this diagram will show that forces in the direction P will cause the two discs to roll together, whilst forces in the direction Q cause no rotation whatever. Forces perpendicular to the paper can, of course, produce no rotation of the disc, so that the instrument records the acceleration in one of the three directions of space only, and is not affected by whatever may be happening in the other two. Thus the dial may even be tilted until it is vertical, so that the whole force of gravity acts across it without the readings being affected. The author believes that this is the first accelerometer instrument to read in one selected direction only.

It is evident that when the instrument is tilted in the direction of motion the needle will move through an angle corresponding to the angle of tilt, and this angle can be read from the (red) graduations on the dial. The instrument thus becomes a "gradometer," and it has the property that it can be used just as well for this purpose on a car which is climbing a slope as on one which is standing on the slope, always provided that the speed of the car is uniform. Although the instrument is thus sensitive to slopes, it is the fact that the tractive resistance readings do not need correction on account of the measure-

ments having been made up or down hill. This may appear at first sight surprising. The explanation is, however, that when coasting down a hill the slope tends to throw the disc forward, whilst the gravitational acceleration due to the slope throws it in the opposite direction, and by just an equivalent amount, so that the two effects neutralise one another; and were there no tractive resistance whatever the instrument would indicate zero. What, in practice, causes the needle to move away from the zero mark is the frictional resistance due to motion. The needle, therefore, shows the amount of this resistance, no matter whether the motion of the car be on a sloping road or not.

As a matter of practical convenience it is best to make such measurements when on a slight downward slope between 1 in 30 and 1 in 60. The reason for this is that the speed then changes more slowly, so giving a longer time for taking the readings, and the complicating effect —familiar to railway engineers—of the change in the momentum stored in rotating parts is then insignificantly small. It may be mentioned that if the rotational momentum[1] is *c* per cent. of the translational momentum, the true tractive resistance is *c* per cent. higher than that deduced when coasting on a *level* road from the negative acceleration shown on the dial of the accelerometer. When tests are made on a road which is not level the instrument reading shows the acceleration, or retardation directly due either to the engine, to the brakes or to the tractive resistance (or a combination of them), and in all cases where these measurements are being made a knowledge of the slope is unnecessary.

To prove this, let the downward slope of the hill be S (so that for a grade of 1 in 10, $S = \frac{1}{10}$), the

[1] See Appendix I.

tractive resistance be R (lb. per ton), and the acceleration due to gravity be A (feet per second per second), the clutch being out. Then the needle of the instrument tends, by reason of the slope, to rest at a reading equal to 2,240S lb. per ton; and by reason of the acceleration to move in the opposite direction by an amount 70A lb. per ton,[1] so that the net reading will, so far as these factors are concerned, be (2,240S — 70A) lb. per ton. We have now to take account of the tractive resistance retarding the motion of the car—its effect will be in the opposite direction to the acceleration, and will in amount be R lb. per ton. Taking this into account we have the net result

$$(2{,}240S - 70A + R) \text{ lb. per ton.}$$

Now A, the acceleration due to gravity, will be ($g \times S$) feet per second per second, equivalent to ($70gS$) lb. per ton, or 2,240S lb. per ton, so that the expression (2,240S — 70A) equals zero and the instrument reading is R lb. per ton, showing that we obtain in this way the value of the tractive resistance, irrespective of the slope on which the measurement is made.

If, however, either the engine is working, or the brakes are on, or both, so that the net effect is to produce a certain tractive effort—positive or negative—in excess (in the positive sense) of the tractive resistance at the moment, the instrument will indicate this excess whatever the hill slope may be. In that case we may put E for the excess tractive effort lb. per ton, and the net reading =

$$= (E + R) - (2{,}240S - 70A + R) \text{ lb. per ton}$$
$$= E + R - R = E \text{ lb. per ton.}$$

And this surplus divided by 2,240 gives the gradient up which this engine effort could just take the car.

This fact it is that enables a reading to be at once obtained of the hill-climbing ability of any car. If the slope the car is on happens to be this slope, the speedo-

[1] This 70 is got by dividing 2,240 by 32, the gravitational constant.

meter will show a perfectly steady speed, and the needle then indicates the hill slope. The car may, of course, also be intentionally driven so that the speed is steady and a reading of the gradient be obtained without stopping the vehicle. When the speed is not steady but is increasing or decreasing, the needle then indicates a slope steeper or less steep than the one the car is climbing. But in either case the following rule holds good :—

The needle points to the gradient which the car could just climb steadily, on that particular surface with that amount of throttle opening and at the speed in question.

This is readily seen to be the case. If the car be on a slope less steep than the one it could climb steadily the speed will accelerate. The needle will therefore be thrown across the dial by an amount equivalent to that acceleration, and as this will be in addition to the swing of the needle due to the hill a total effect will be recorded which will be equal to the sum of the two, *i.e.*, to the total tractive effort that the car can exert at that speed. And this may be read on the dial in feet per second per second, or in gradient as may be desired.

Thus, if a car can just climb steadily a gradient of 1 in 10 at a given speed, and if it be set to climb a gradient of, say, 1 in 14, there will be an excess of tractive effort of $(224 - 160) = 64$ lb. per ton. And this will, of course, produce an acceleration of $64 \div 70 = 0{\cdot}91$ feet per second per second. Owing to the slope the needle tends to point to 1 in 14 (*i.e.*, $\frac{g}{14}$ or 2·31 feet per second per second), but the acceleration of 0·91 carries it on to a total reading of $(2{\cdot}31 + 0{\cdot}91) = 3{\cdot}22$ feet per second per second, equivalent to a gradient of 1 in 10. The same argument applies to hills steeper than the critical slope. In all cases the instrument reads the critical slope no matter what the road conditions may be.

In much the same way it can be shown that when the engine power is cut off and the brakes are applied (singly or together), the needle points to the gradient of the steepest hill on which that amount of braking would just hold in the car. And this reading is the same whether the road on which the experiment is made be level or not. Therefore it is not necessary to know the road slope.

It is important always to bear in mind that when a test is to be made with this instrument with the object either of ascertaining the hill-climbing ability on each gear or finding the braking power of each of the brakes, there is no need to concern oneself with the road slope on which the tests are made. The instrument itself makes the necessary corrections. As an illustration of the various modes of using this instrument the author had occasion to seek to measure the hill-climbing ability of a certain vehicle on bottom gear. The country all round was exceedingly flat, so that the usual method was not at the moment available. He therefore caused the car to be driven on bottom gear on a level stretch with the hand brake just on hard enough to keep the speed steady at the probable figure for hill climbing—of course, this was only continued for a few yards—then the engine was declutched, and the accelerometer at once indicated the total resistance, which was 300 lb. per ton, showing that a hill of 1 in 9 could be climbed with a road resistance of 50 lb. per ton. This is not a very good method when a hill of some kind can be found since the car comes so rapidly to rest that there is little time in which to make the measurement. But it illustrates very well what can be done by varying the conditions of running and making accelerometer measurements.

Resistance Measurements—Numerous tests of tractive resistance have been made. Some early ones, taken

on a heavy, slow-speed, touring car, are shown in Fig. 8. The rise of resistance[1] at higher speeds due to the growing importance of air resistance is clearly seen. In the table below are given a number of tractive resistance figures for motor wagons at low speeds on various road surfaces. Even with motor wagons the air resistance term is appreciable,

Name and Condition of Road.	Tractive Resistance (declutched), lb. per ton.	Tires.
Tar, macadam, hard	65	Steel behind, solid rubber in front
Ditto, soft and cut-up... ...	140	Ditto
Granite setts...	50	Ditto
Clean wood pavement with tram lines	70	Ditto
Tar macadam dry and hard	70	Ditto
Ditto, very muddy and sticky...	95	Ditto
Road metal, partly rolled ...	120	Ditto
Ditto, unrolled	200	Ditto

since although the speed of motion may be low, the windage area is considerable.

It was unexpected at first that a wagon running at 15 miles per hour would be appreciably affected by air resistance, and the rise was attributed to some unknown

[1] The following extract from Strickland's "Petrol Motors and Motor Cars" states (as a result of draw-bar measurements) the assumptions formerly made as to the value of that part of the tractive resistance which did not depend on air resistance :—"On an ordinary road it is "between 40 and 100 lb. per ton according to whether the surface is "dry or muddy, etc. In the small car trials in 1904, the Committee of "the Automobile Club estimated the road resistance at 60 lb. per ton. "Probably a road resistance of 50 lb. per ton for a pretty good road "may be taken as a safe basis for a calculation." It will be noticed that this assumes that the whole of the V^2 term is due to air and air only.

cause. Part of the rise is, no doubt, due to a term proportional to the speed and corresponding to the loss of energy through the bumpiness of even the best roads. The most satisfactory formulæ to suit these curves are, however, of two terms only—a constant term and a term proportional to the velocity squared.

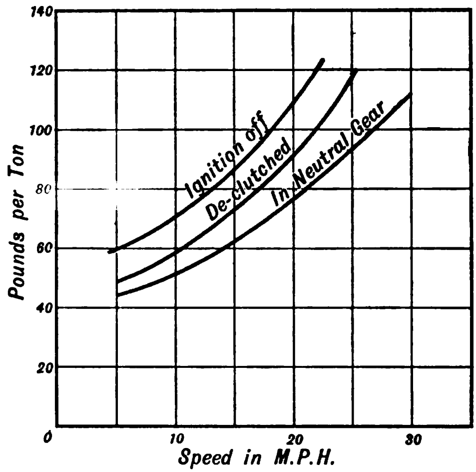

Fig. 8.—Tractive Resistance Curves obtained with Accelerometer.

For the touring car—from Fig. 8—

$$R = 47 + 11{\cdot}5\left(\frac{V}{10}\right)^2$$

This is for a car fitted with a canopy and having solid tires. For most ordinary touring cars with the hood down and fitted with pneumatic tires, the following formula fits very nearly—

$$R = 50 + 6\left(\frac{V}{10}\right)^2$$

One often sees air resistance formulæ given in terms of the " projected area " of the car and of the velocity. It is, however, difficult to say what the true resistance area

really is. In the case of the canopied touring car the projected area of the front of the car was twenty-eight square feet, which on the basis of the usual air-pressure formula $0{\cdot}003V^2A$ would give an absurdly low result compared with that obtained by experiment.

RESISTANCE OF MECHANISM—It has already been established by the investigations of several observers that engine friction is independent of the load and dependent on the speed. It is important to enquire how it varies with the speed of rotation. The dependence of the resistance on engine speeds arises from the fact that the rubbing surfaces are lubricated, and hence the laws of fluid friction in part apply. The differences in the heights of the two upper curves shown in Fig. 8 and of similar curves taken on a motor wagon, are shown plotted in the upper part of Fig. 9. This gives the engine friction, which is seen to follow some such law as

$$R = 10 + 10\left(\frac{\text{r.p.m.}}{1{,}000}\right) \text{ lb. per ton.}$$

The author has not found any measurable difference in this resistance due to changes in position of the throttle lever.

In the lower half of Fig. 9 is seen the result of plotting the differences in heights of the lower pairs of resistance curves. This shows the effort necessary to rotate the inner part of the clutch with the shaft and gearing connected to it. The great rise of resistance with speed is remarkable, and the following formula is approximately followed :—

$$R = 5 + 12\left(\frac{\text{r.p.m.}}{1{,}000}\right)^2 \text{ lb. per ton.}$$

There is no doubt that the reason for this increasing resistance lies in the rapid rotation of the change-speed wheels in a gear-box filled with lubricant. A consequence is that when running down hill vehicles can run much more freely in the neutral gear than when

declutched. Fig. 8 shows that when running steadily down a slope of 1 in 20 the speed, when declutched, would be 24 miles per hour, and when in the neutral gear 6 miles per hour faster.

Transmission Losses—It will be remembered that the H.P. absorbed in rotating the engine is almost con-

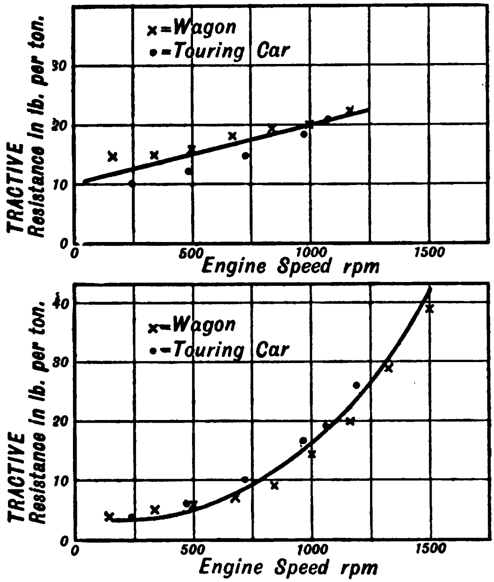

Fig. 9.—Engine Friction (above), Gearbox Friction (below), deduced from curves similar to Fig. 8.

stant at all loads provided the speed is constant (indeed, it has become customary even in scientific tests to obtain I.H.P. by adding to the B.H.P. the H.P. needed to turn the engine round light). This means that the resisting torque due to friction depends only on the speed and not on the load. We have seen that it rises rapidly with speed and we therefore know that it follows the laws of fluid friction much more nearly than those of solid.

This arises, of course, from the fact that the cylinder walls and piston are lubricated. It may naturally be expected, therefore, that other parts of the mechanism which are lubricated to at least an equal degree will exhibit something of the same phenomenon. It would, in fact, be reasonable to expect that gear box friction and the various bearing frictions would all of them be very dependent on speed but comparatively little dependent on the power passing. If this were so it would mean that the power lost in transmission would be about the same at say 20 m.p.h. whether the car were running on a level road or up a considerable gradient, although the hysteresis loss in the rubber tires would of course be greater.[1]

When we measure tractive resistance with an accelerometer, we include in it the internal resistances of the car between the road wheels and the clutch. If these resistances were materially greater when the engine was working than when it was not, the tractive resistance figure so obtained would be an underestimate and the deduced B.H.P. would be low. Bench tests of gear box losses under varying loads and speeds would show how far this is the case, but few such measurements have been made, and it is necessary to check the work in another way. Now the accelerometer method of obtaining B.H.P. at full speed on the level should check against the bench test B.H.P. of the same engine at the same engine speed. A number of cars of different makes have been carefully examined in this way and the concordance of the results —which agree within 3 per cent.—shows that the assumption made is, within these limits, as correct for the transmission mechanism or direct drive as it was previously known to be for the engine mechanism. This

[1] NOTE.—For ordinary running the horizontal reaction at the tire is small compared with the vertical reaction, but when climbing a steep hill the horizontal reaction is of the same order of magnitude as the vertical and the additional hysteresis is noticeable.

result is not unexpected, since the greater part of the transmission loss at high car speeds is due to the churning of the lubricant in the gear box (which is most often more than half full).

From this it follows that such road tests of engine B.H.P. are as reliable as those on the bench, and they have the advantage of being carried out under a close approach to real working conditions. In fact, it is possible with a little care to reproduce the complete engine torque curve.

MEASUREMENT OF GEAR BOX LOSS—In a paper read by Mr. L. A. Legros before the I.A.E. in 1908-9, some tests made in America by Mr. Henry Hess were quoted. These figures are reproduced below, but it is uncertain whether the temperature of the lubricant was kept constant throughout, and the irregularity of the figures was pointed out by Mr. F. Strickland during the discussion of Mr. Legros' paper.

Nevertheless, the figures are worth reproducing as conveying some idea of the extent of the loss. The very large loss on the "reverse" is due to the larger number of wheels then in mesh.

AMERICAN EXPERIMENTS—QUOTED IN MR. L. A. LEGROS' PAPER IN "PROC. I.A.E.," IN 1908–9—BY HENRY HESS ON A GEAR-BOX WITH PLAIN BEARINGS, THREE SPEEDS AND REVERSE WITH DIRECT DRIVE ON TOP GEAR. ("MOTOR TRADER," SEPTEMBER 25TH, 1907.) SPEED ABOUT 1,200 R.P.M. IN ENGINE.

E.H.P. supplied to motor.	Equivalent B.H.P.	Transmitted B.H.P.				Efficiency per cent.			
		3rd.	2nd.	1st.	Reverse.	3rd.	2nd.	1st.	Reverse.
7	5·90	5·20	5·12	4·86	—	88·1	86·8	82·4	—
9	7·65	6·94	6·68	6·44	5·98	90·7	87·3	84·2	78·2
11	9·44	8·50	8·21	7·92	6·80	90·0	87·0	83·9	72·0
13	11·19	9·95	9·64	9·35	7·48	89·8	87·0	84·3	67·5
15	12·55	11·34	11·00	10·60	8·02	90·4	87·6	84·5	63·9
17	13·89	12·65	—	—	8·55	91·0	—	—	61·6
19	14·90	—	—	—	9·02	—	—	—	60·5

BEVEL AND WORM GEAR LOSS—Little reliable has been published under this heading. What is published is usually suspiciously favourable to the gear preferred by the experimenter. But the following account of some American experiments which seem to have been carefully carried out is worth giving:—

Tests[1] were made on three forms of worm gear and a bevel gear, the apparatus being as follows. An electric motor drives, through an automobile gear-case, a transmission dynamometer which in turn drives a friction brake through the worm and worm wheel. The worm was covered with lubricant in an oil-bath, and the temperature of the oil was noted. The efficiencies were found to increase with the power transmitted; with the worm running at 880 r.p.m. and its wheel at 198 r.p.m. and transmitting 30, 40, and 50 H.P., the worm gear efficiency was 92·4 per cent., 94·4 per cent., and 97·9 per cent. respectively. The bevel gear used for purposes of comparison had a ratio of 14 to 52 and the efficiency was found to be 94·2, 95·1, and 98·9 at 30, 40, and 50 H.P. respectively.

The National Physical Laboratory has carried out some very valuable tests on the efficiency of worm gearing[2] made by the Daimler Co. The worm in question was what is known as the "hollow" type, to distinguish it from the "parallel" worm. Illustrations of the two types are seen in Fig. 9A. The testing machine used by the National Physical Laboratory was built by the Daimler Co. to the designs of Mr. F. W. Lanchester, and it has also been used at the Daimler works to test the efficiency of the "parallel" type of worm in comparison with the "hollow" type. The following table gives the results.

[1] W. H. Kenerson, Am. Soc. M.E., September, 1912.
[2] F. W. Lanchester in *Proc. I.A.E.*, 1913.

R.P.M.	Efficiencies of Worms (per cent.).			
	"Parallel."		"Hollow."	
	A.	B.	A.	B.
500	93	91	95	94
1,000	94	92½	97	95½
1,500	94½	92½	96	95½

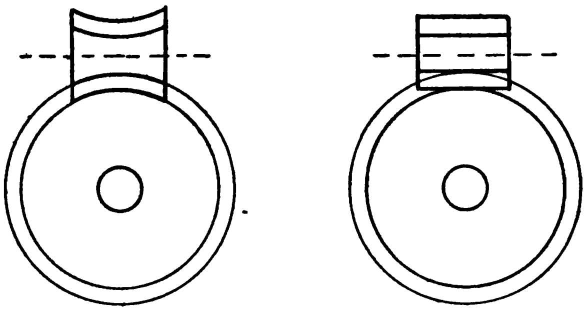

FIG. 9A.—Diagrams of the two forms of Worm.

In case A the pressure was 1 ton per square inch of projected area of tooth, and in case B, 2½ tons. It will be seen that the hollow worm is from 1½ to 3 per cent. more efficient than the parallel worm, and that no great error will arise if the loss in the hollow worm be taken at the average figure of 5 per cent.

Mr. Lanchester compares his results with some bevel gear efficiencies determined by Mr. Faroux, who found the following results.

R.P.M.	Bevel Gear Efficiency (per cent.).
500	88
1,000	87
1,500	66

But there must have been something exceptional in these tests since the bevel gear efficiency would not be as low. The testing machine, it may be noted, was greatly inferior in accuracy to that used by Mr. Lanchester.

The following table of Mr. Lanchester's shows the effect of varying the pressure on the teeth of hollow worms of different gear ratios and at different speeds.

EFFECT OF PRESSURE ON TEETH ON EFFICIENCY AT 30°-40° C.

Gear ratio.	Pressure on Teeth. Lbs. per sq. in.	Efficiency (per cent.) at			
		1,500	1,000	700	400 R.P.M.
8/33	682	93·7	93·5	93·7	93·6
	1,205	95·4	95·2	95·1	93·9
	1,733	95·7	95·2	95·3	94·0
	2,258	95·65	95·5	95·1	93·8
	2,786	95·6	95·5	95·2	93·7
8/35	1,200	95·65	96·8	95·4	94·6
	1,727	96·1	95·7	95·4	95·1
	2,780	95·75	95·4	95·2	93·1
9/34	1,200	96·0	96·3	95·7	95·2
	1,727	96·45	96·1	95·8	95·0
	2,780	96·7	96·3	96·1	95·0

For touring cars there is no doubt that the worm drive (whether parallel or hollow) is quite satisfactory. For heavy vehicles, however, it is still on its trial, since the exceedingly heavy torques to be transmitted are liable to squeeze the lubricant out. Perhaps the "hollow" worm may have a successful field in this work. In Mr. Lanchester's opinion the "hollow" worm "will easily carry one ton per square inch, and is good for an overload of two or three times that amount, in fact, a load of two tons per square inch may be looked upon as a

safe load, inasmuch as the gears will run satisfactorily with such a load for an indefinite period."

ROLLER AND BALL-BEARINGS—The frictions of ball and roller-bearings vary in much the same way, although the former is somewhat lower in amount, and each is, of course, much less than that of plain bearings. Plain bearings require far more attention than ball or roller-bearings, and to an increasing extent they have been replaced by the latter in recent years for motor vehicle work.

Prof. Goodman[1] found that the coefficient of friction of ball-bearings with flat races, decreases (and with grooved races sometimes increases) with increase of load, but it is much more nearly constant than that of plain lubricated bearings. He also found the ball-bearing friction to be practically constant at all speeds, but to have a slight tendency to decrease with increase of speed. The coefficient is independent of the temperature—which in plain lubricated bearings is far from being the case. The same series of tests showed that the starting effort with ball-bearings is practically the same as the running friction, and that the effect of lubrication on a well-designed ball-bearing is to *increase* the friction and not to diminish it.

FUEL ECONOMY—It is very usual to hear statements made of the miles run per gallon of fuel, or of the "ton-miles per gallon." Since the former ratio will obviously vary with the weight of the vehicle, it is better to take weight into account as is done in the expression ton-miles. A slight ambiguity exists here since some persons take the tonnage carried on the vehicle instead of the total weight of the vehicle plus its load. The former method may be useful for commercial purposes, but as a test of the vehicle the latter is far preferable. To make one's meaning clear it is usual, therefore, to speak of

[1] *Proc. I.C.E.*, Vol. 189.

gross-ton-miles per gallon, or more briefly, G.T.M. per gallon. A common figure for this, when petrol is used, is about 50.

Now it happens that the G.T.M. per gallon is really a measure of thermal efficiency. Thus, if the tractive resistance be 70 lb. per ton, and the energy stored in each gallon of petrol be 100,000,000 ft. lb., then 50 G.T.M. per gallon of petrol means that the vehicle has performed work equal to moving 50 tons a distance of one mile against a resistance of 70 lb. per ton, with a consumption of thermal energy equal to 100,000,000 ft. lb.

To move 50 tons a mile against this resistance involves the expenditure of work equal to

$$50 \times 5{,}280 \times 70 \text{ ft. lb.} = 18{,}500{,}000 \text{ ft. lb.}$$

So for every 100 ft. lb. of thermal energy, 18½ ft. lb. of mechanical work are done, showing that the brake thermal efficiency of the engine is 18½ per cent.

With light vehicles the G.T.M. per gallon figure tends to be lower than with heavy, since the former are usually speedier and the tractive resistance is consequently higher. If all engines were equally efficient then the product of G.T.M. per gallon by tractive resistance would be constant.

A test, therefore, which shows a low figure for the G.T.M. per gallon does not, of necessity, mean a poor engine. It may merely mean that the average speed was high, or that there was a head wind acting against the vehicle all the time, or that the roads were unusually heavy. In such cases it is most advisable also to record the average tractive resistance and to make a proportional allowance in the figure for G.T.M. per gallon.

If, for instance, two similar vehicles were tested on two different days and gave 60 G.T.M. and 35 G.T.M. per gallon respectively it would not necessarily follow that the second was the worse machine. If the tractive

resistance on the day of the first test were 50 lb. per ton, and on the day of the second 95 lb. per ton, then, reducing both to the common basis of 70 lb. per ton, the two figures would be respectively $60 \times \frac{50}{70}$ and $85 \times \frac{95}{70}$, or 43 and 47 G.T.M., showing that the second was really the better performance.

A certain car tested by the R.A.C.[1] in April, 1913, had its fuel consumption measured at a number of different speeds. At the higher speeds when the tractive resistance would be greater, the ton-mileage per gallon of fuel was less. The following table taken from the report shows this clearly:—

Speed. m.p.h.	Miles run per gallon.	Gross ton-miles per gallon.
13·35	33·40	62·34
19·31	30·44	56·82
25·10	28·63	53·44
31·12	26·52	49·51
37·17	21·67	40·44

[1] R. A. C. Journal, 2nd May, 1913.

CHAPTER III

General types of steam and internal combustion engines and of vehicles on which they are used—H.P. per ton—Use of gearing—Ideal tractive effort curve—Loads—Gradients

Chassis Arrangement—The power necessary for the propulsion of road motor vehicles may be derived from steam engines, internal combustion engines, or from electric motors. The H.P. per vehicle may range from 5 to 100, but is generally between 10 and 40. When exerting anything approaching full power the speed of the motor shaft will usually be over 1,000 r.p.m., and not uncommonly very much more. But the speed of the road wheels will always be much less and some form of gearing to connect the two is necessary.

A diagrammatic view is given in Fig. 10 of a common chassis arrangement for a motor car, and is much the same in principle even when a steam engine or an electric motor is used; though in the case of the steam engine we have greater flexibility of engine speed, and in the latter the motor speed is usually controlled electrically so that no gear box is needed. The diagram shows a bevel drive at the back axle, and this is still the most common arrangement. It is, however, often replaced by a worm and worm wheel. The latter arrangement may be less efficient than the bevel, particularly at heavy loads, but it has the advantage of being silent in action, and it has grown in popularity in recent years. Sometimes the bevel or worm gear is not mounted on the back axle but on a kind of countershaft, known as

a cross-shaft, from which the driving effort is transmitted to the back shaft by a pair of chains. This plan is often followed on very heavy vehicles in which the speed of motion is low. In all cases the worm or bevel gear contains a "differential," so that the two road wheels may run at different speeds when rounding corners. The effect of the differential is to divide the back axle torque equally between the two road wheels whatever the angular velocities of the two wheels may be.

The fly-wheel and clutch are very commonly made in one. The fly-wheel is needed to steady the rotation of

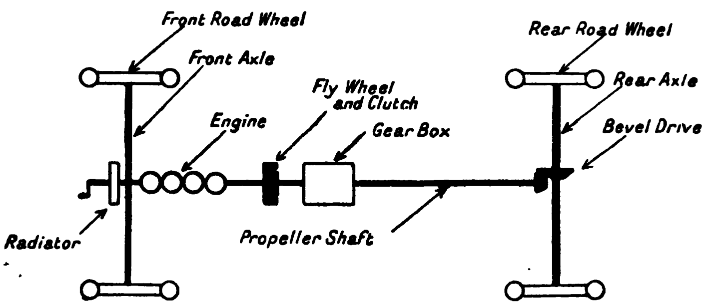

FIG. 10.—Outline of typical Motor Chassis.

the engine and to store up energy when the driver wishes to start the vehicle from rest. The clutch should be so designed that as it is let " in " it gradually transmits the energy from the fly-wheel and engine to the whole mass of the vehicle. In this way the vehicle gains speed until the engine is able to take up the load. There are many types of clutch in use. The commonest is the cone clutch, having a leather or ferodo face. Plate clutches are used when a very gradual starting effort is needed.

The gear box is a device for altering the gear-ratio between engine and road wheels, so that it may be possible to reduce the range of speed of the engine within

small limits, even though the speed of the car may vary between exceedingly wide ones. Usually it is so arranged that on " top speed " the drive is " direct," and that on other speeds a lay-shaft, like that on the back gear of a lathe, is brought into use. Sometimes an electrical arrangement replaces the gear box described, but this is not yet in general use. At the driver's end of the gear box is the shaft known as the " propeller-shaft " ; this has a universal joint at each end to allow the rear axle to rise and fall.

A photograph of an actual chassis is seen in Fig. 11. The propeller-shaft is here seen to be enclosed in a strong casing which acts also as a " torque-rod " in preventing the engine torque from distorting the transmission gear. One of the radius rods may also be seen. These rods transmit the forward drive of the rear wheels to the chassis framework. In some designs the rear springs are made to act also as torque rods and sometimes as radius rods too. In the design shown, however, the preferable plan is followed of having separate parts for these functions, so that the springs are free to do their own work and that only. Other views of this same chassis are shown in Figs. 12, 13 and 14.

ENGINES—Steam engines are generally double acting, and where there are two cylinders compounding is very usual. The working of a steam engine and its slide valve is well known, and much the same now applies to the internal combustion engine, in which, however, it is necessary to distinguish between four-stroke engines and two-stroke engines.

The four strokes of the four-stroke cycle were shown in Fig. 3, which is here reproduced for convenience (see Fig. 15). It is seen that at the end of the suction stroke the cylinder is filled at little below atmospheric pressure with the working charge for the next stroke.

It is also seen that during the expansion stroke we have expansion occurring to the very end of the stroke.

Now in the two-stroke engine neither of these processes is so well carried out. The cylinder does not get so completely filled nor does the expansion continue so far. So that although for the same engine speed there are twice as many working strokes in the two-stroke engine as there are in the four-stroke, the power of each stroke is not so great, so that some of the gain is

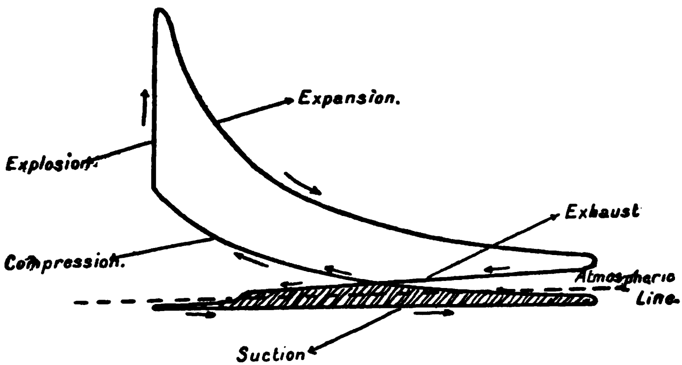

FIG. 15.—Indicator Card for Four-stroke-cycle Internal Combustion Engine.

lost. The way in which this loss occurs in a common form of these engines is seen from Fig. 16. As the piston gets near the end of the expansion stroke it uncovers the exhaust part shown on the left, and the burnt products escape—thus expansion stops short of the end of the expansion stroke. A very little later in the stroke the port on the right is uncovered, and a charge of air and gas is forced in from the crank case by the displacement of the descending piston. This is then compressed by the upward motion of the piston, whilst a fresh supply of air and gas is drawn into the crank chamber by the

uncovering of the lower port shown on the left. But the compression pressure is not so well reached as in the four-stroke engine, and at the beginning of the stroke some gas escapes across the piston face, in spite of the baffle shown, and out at the exhaust port. Also it is not easy to keep the crank case air tight and to prevent back-firing into it.

Taking everything into account, therefore, there has

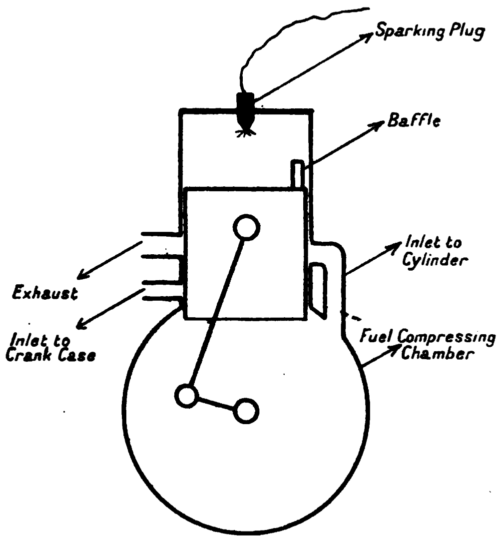

FIG. 16.—A common form of Two-stroke Engine.

been little gained hitherto by the use of the two-stroke engine. Mr. Ralph Lucas,[1] who has done much work at this subject, has expressed his view that the mean pressure developed in a two-stroke motor car engine is only about half that in a four-stroke engine.

There is another way of looking at this matter. The object of the designer of a two-stroke engine is to get, if

[1] "Report of Treasury Committee on H.P. Rating of Motor Cars, 1912." (Cd. 6415.)

possible, as strong a working stroke as in the four-stroke engine, and as a consequence twice the power, for given engine dimensions. Now if the power is to be doubled in this way, double the fuel must be used. This means that the heat units received by the engine must also be doubled. And if the heat units received and used are doubled those wasted will also be doubled. The heat loss to the walls must be doubled, therefore, and this means that the average temperature gradient in the walls will be twice as steep. Thus we see that a special design of engine is needed to avoid troubles due to overheating and perhaps cracking of cylinder walls or heads. It is not as easy to design a two-stroke engine as may at first sight appear. In some cases it becomes more complicated than the four-stroke engine, and engineers are unwilling to pay this price unless the balancing advantages are adequate—and it is often found that the advantages gained are no proper balance to the complication involved. The action of the four-stroke engine is exceedingly simple, and the compression of the charge is very efficiently performed. Simplicity is, however, achieved in a very remarkable way in one of the two-stroke engines, viz. the Ricardo.

Definition of the H.P. of a Motor Car—The H.P. of any engine[1] or motor depends entirely on the conditions under which it is measured, and these conditions must be specified, either expressly or by implication, in the statement of H.P. if that statement is to have any meaning. The necessity for arriving at clear ideas on this point before proceeding to discuss the rating of motor cars may be illustrated by some figures based upon tests. A certain car, described by the makers as of 12—16 H.P., was capable of doing a maximum speed of thirty-two

[1] "Report of Treasury Committee on H.P. Rating of Motor Cars, 1912."

miles per hour on a good level track and then the engine developed 19 H.P. About the same, or possibly rather greater, power could be produced in the bench test of the engine at the maker's works, where it would be run for an hour or two under a constant load equal to, or near, the maximum which it could sustain for such a period. Driven on a road, under the restrictions on speed imposed by traffic conditions, this car could, in regular use, maintain an average speed of perhaps twenty-two miles per hour. The H.P. in such a run would, of course, vary at different times from nothing, when running down-hill, up to the maximum of perhaps twenty, obtained for short periods when accelerating or going up-hill, but the average would be approximately that corresponding to twenty-two miles per hour on a level road, and would be about nine H.P. This may be called the touring H.P., and corresponds to a speed determined mainly by traffic restrictions. If these were removed, the average speed would be limited by the requirements that the engine must not be worked so hard as to cause it to break down or to wear out unduly fast, and that the occupants must not suffer discomfort from vibration. These requirements would in general fix it at a figure higher than the touring speed, but considerably less than the racing speed, and there would be a corresponding value of the H.P.

It will be seen that there are at least three different meanings which may be attached to the H.P. of a car, the values ranging in the particular case considered from 9 to 20, and any one of these might be defended on various grounds as a possible basis of rating.

PETROL ENGINE RATING—The rating adopted by the R.A.C., and now confirmed by the Treasury Committee, was originally suggested by Dr. Dugald Clerk. He proposed that

$$\text{Rated H.P.} = nd^2 \div 2{\cdot}5,$$

where n is the number of cylinders and d the cylinder diameter in inches. Thus a four-cylinder engine of 4-inch bore would have a rated H.P. of $(4 \times 16) \div 2{\cdot}5$ or $25{\cdot}6$ H.P.

Now this is tantamount to assuming that for a given mean pressure in the cylinder the piston speed in feet per minute will be the same for all types and sizes of engine. Experiment has shown that the mean pressure is practically independent of bore and stroke, but there is uncertainty as to how far it is safe to assume that the piston speed is the same in all engines. This uncertainty is due to the lack of clearness as to what the rated H.P. is supposed to represent. It might be any one of these:

(1) Maximum H.P. on the bench when running "all out."

(2) Maximum H.P. on level road when running "all out."

(3) Maximum H.P. when climbing the steepest hill climbable.

(4) Average H.P. when running on roads in normal unimpeded service.

Now for (1) the R.A.C. rating method is probably the most correct, using, however, a lower constant than 2·5. For (2) the engine speed in r.p.m., and not the piston speed, is the more nearly constant thing, and H.P. is therefore roughly proportional to cylinder volume, and roughly equal to volume in cubic centimetres $\div$ 100. For (3) the engine speed will always be about 800 r.p.m., and the volumetric rating applies here also, and we may roughly put $\text{H.P.} = \dfrac{\text{displacement in cubic centimetres.}}{130}$ For (4) there is little data available, but the R.A.C. formula probably fits very nearly with its present constant. We may therefore make out the following table:—

d = cylinder diameter in inches. n = No. of cylinders.
C = total displacement volume in cubic centimetres.

Engine B.H.P. =

(1) Maximum bench H.P.	$= knd^2$ where k is some constant.
(2) Maximum H.P. on road	$= C \div 100$
(3) Maximum H.P. when climbing steepest hill possible	$= C \div 130$
(4) Average H.P. in uninterrupted service on roads	$= nd^2 \div 2{\cdot}5$

THE RATING OF STEAM CARS [1]—The essential difference between a steam engine and a petrol engine is that whereas in the latter the power depends upon the engine alone, in the former the primary source of power is the boiler. The boiler is capable of supplying steam at a certain average rate, and in the long run the engine can only develop power at a corresponding rate. By drawing on the reserves of steam in the boiler it may produce for short periods an amount of power greatly exceeding the average, but, unless the boiler be a very large one, such periods of over-load will result in a drop of pressure, and must be set off by periods of comparatively light load, during which the boiler can make up the pressure again.

In a steam car, as in other cars, the engine is only occasionally called upon to develop its full power, and the average is much less than the maximum. These short spurts at high pressure can be provided for by a comparatively small boiler—a boiler quite incapable of producing steam at that rate continuously. The maximum power available for short periods—say, in ascending a hill—is determined largely by the size of the engine. But the average power on a long run depends solely on the boiler. The power of a steam car ought to be taken as

[1] "Report of Treasury Committee on H.P. Rating of Motor Cars, 1912."

proportional to the capacity of the boiler for evaporating water, and this depends upon the heating surface, and on the rate at which the burner can consume fuel. The ideal rating formula for steam cars would be based simply on these quantities, and would be independent of engine dimensions.

The capacity of a boiler depends upon the heating surface exposed to the flame and on the arrangements for burning fuel. With equally efficient burners the capacities of different boilers will be in proportion to their effective heating surfaces, and it is not practicable to take account in rating of the nature or size of the burner. The H.P. rating of a steam car must therefore be taken as proportional to the effective heating surface of the boiler. This is not difficult to ascertain in the boilers of the types now used in motor cars. In horizontal tube boilers the whole of this surface may be taken as effective. In vertical tube boilers a portion of the tube surface is not fully effective, and in such cases it would be reasonable to take the effective surface as half the total.

The Treasury Committee considered that three square feet of heating surface should be taken as equivalent to one H.P. This rule, bearing in mind that in a vertical tube boiler only half the heating surface is to be considered effective, will, when applied to the various steam cars now on the market, give H.P. values corresponding sufficiently nearly with the definition of H.P. adopted.

The Rating of Electric Cars—The primary source of power in the electric car is the storage battery, which is to some extent analogous with the boiler of a steam car, the electric motor corresponding to the engine. There is, however, the important difference that whereas the output of the boiler is strictly limited by the heating surface and burner capacity so that it really determines the power of the car, the possible output of electric current from the storage battery is always much in excess of the

rate at which the motor can safely transform it into mechanical work. By putting in a larger battery the car can be made to go longer without re-charging, but whether the battery be large or small it is possible for it to produce current without serious damage to itself at such a rate as will burn out the motor. Hence in the electric car the power is limited by the motor.

The H.P. of an electric motor can be determined experimentally with some precision. When the motor is working continuously under a constant load its temperature rises by a definite amount which depends on and increases with the load. The H.P. is the maximum load corresponding to a safe working temperature. The permissible temperature is the highest consistent with durability and freedom from breakdown; and when this has been fixed the rating of any electric motor is definitely settled. While manufacturers naturally differ to some extent as to the basis of temperature on which they rate their motors, there is fair agreement among the best of them.

The rating of the electric motors in cars may therefore be fixed by reference to temperature, the limit of temperature being chosen so as to be in accordance with the average practice of manufacturers.

H.P. PER TON—An important series of road trials of motor wagons was held in 1907. These trials were organised by the R.A.C., and they showed the following average ratios of rated H.P. to gross moving weight :—

Net Load.	H.P. (R.A.C.) ÷ total weight in tons.
10 cwt.	7·82
1 ton.	6·8
1½ tons.	7·24
2 ,,	4·96
3 ,,	5·01
5 ,,	4·87

These statistics show what was the practice in 1907, but it is not likely that a similar set of figures to-day would be very different. On the whole engines have become more powerful for a given load, but the R.A.C. rating H.P. would not be much, if any more. It will be noticed that the H.P. per ton is higher in the smaller vehicles, which is due, of course, to their higher speed calling for a greater tractive force.

It is seen from this that an average motor wagon calls for about five rated H.P. per ton. With touring cars a very much greater amount of H.P. is needed. Thus, in some accelerometer tests carried out in 1912 at Brooklands at the instance of the R.A.C., the following data were obtained :—

RATED AND MAXIMUM H.P. OF 15 MODERN CARS COMPARED WITH WEIGHT.

H.P. (R.A.C.)	B.H.P. Maximum.	Weight in tons.	Rated H.P. per ton.
12·1	20	1·30	9·3
12·1	26	1·20	10·1
13·9	18½	1·08	12·8
13·9	18	1·21	11·5
13·9	28½	1·26	11·0
15·5	24	1·36	11·4
15·9	19	1·14	11·0
15·9	29	1·20	13·2
15·9	18	1·35	11·7
19·6	24½	1·62	12·1
20·1	33½	1·67	12·0
24·8	24½	1·88	13·2
24·8	22	1·91	13·0
25·8	33	1·67	15·5
25·8	43½	1·98	13·0

This table shows that the average rated H.P. per ton in the case of touring cars is over ten and nearer twelve, *i.e.*, twice as high as in the case of motor wagons. The

maximum B.H.P.,[1] as distinct from the rated H.P., would certainly be much more than twice as high, probably three times. This is because the higher rate of rotation of the engine in the touring car enables the rating to be more easily exceeded. We may, however, take it roughly that five rated H.P. are needed per ton for motor wagons, and twice as much for motor cars.

IDEAL CURVE FOR TRACTIVE EFFORT—Having provided the requisite amount of H.P. per ton of total moving weight, the next important matter is the nature of the relationship between tractive effort and speed. Existing gear-changing mechanisms are none of them perfect from this point of view, and we wish to consider what is the ideal to be aimed at. When the aim is known it will be easier to see how nearly any particular device meets the requirements.

If there were no need to concern oneself with the imperfections of the gear-changing mechanism, all attention would be concentrated in the engine. The limit to the working of the engine is overheating trouble. A given design will only allow of the getting rid of a certain amount of the *rejected* heat without the engine getting too hot. This limits the amount of H.P. that can be developed, since the rejected heat rises with the heat usefully used. There is, therefore, a maximum H.P. at which the engine can be continuously worked.

Now H.P. = tractive force × speed × constant.

So for constant H.P. the tractive force and the car speed will be inversely proportional. That is to say, a graph of the two will be a rectangular hyperbola. This, therefore, is the ideal curve. It requires, however, one amendment. At very low speeds it shows a very great tractive effort. But anything in the neighbourhood of 700 lb. per

[1] NOTE.—It is commonly stated that the B.H.P. falls by three per cent. for each thousand feet of elevation above sea-level, but the author knows of no experiments in support of this.

ton will most often cause the driving wheels to slip. So there is no good providing a greater amount of tractive effort than corresponds to this practical limit. This makes the curve assume the form shown in Fig. 17.

If, now, the internal combustion engine gave a steadily increasing torque with falling speed we could at once attain this ideal. Unfortunately, however, it does not do so. While the engine speed falls from 2,000 to 1,000 r.p.m. the torque does usually increase in a curve not exceedingly remote from this, but a decrease of speed below 1,000 r.p.m. brings one to a torque which, instead of rising, remains almost constant, and on a further lowering of speed it decreases and the engine stops unless some of the load is immediately removed. In practice, therefore, one can only use that part of the torque curve which approximates to the corresponding part of the hyperbola; to build up the rest of the hyperbola three or four other gears have to be used—and even then it is a somewhat patch-work affair, regarded from the standpoint of the ideal. A mechanism which enabled the engine always to work at its maximum safe output, and maintained the gear ratio from crank shaft to road wheels as always inversely proportional to car speed, in other words an infinitely variable transmission gear, would solve the difficulty. But no one has yet invented this. The nearest approaches to it are the electric and hydraulic mechanisms to be discussed later. The irregular nature of

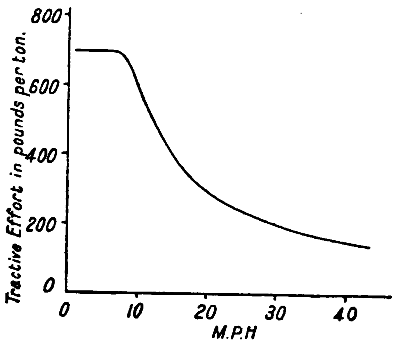

FIG. 17.—Curve of Ideal Tractive Effort.

the starting acceleration with the usual gear box arrangement is seen in Fig. 18. Of course, on very small cars the "friction disc" method of gear changing is excellent, but it cannot be used on heavy vehicles on account of the friction wheel slipping unless a very great force holds it up to the disc, and such a force would be difficult to provide. The immense majority of automobiles use gear wheels for gear changing, and despite the theoretical

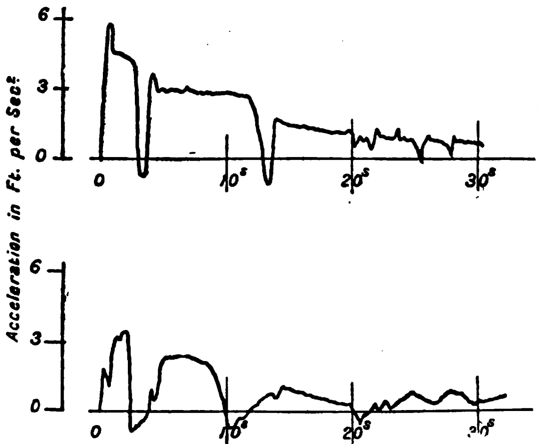

FIG. 18.—Acceleration Curves taken with a Recording Accelerometer. The upper one shows a rapid start and the lower one an ordinary start.

weakness of this method it has been brought to a considerable degree of mechanical perfection. It will, therefore, be necessary in a later lecture to consider how far this means of gear changing can, by careful design, present as many advantages and as few disadvantages as possible.

GENERAL CONSIDERATIONS—We have seen that for slow-moving vehicles, such as motor wagons, about five rated H.P. is needed per ton of total moving weight, and on touring vehicles about twice this amount. It is now necessary to consider how this power is to be provided.

There are many ways possible, and in order to choose between them it is necessary to take many factors into account, such as—

(1) Load to be carried.
(2) Hilliness of the route.
(3) Distance apart of supply depôts.
(4) Nature of fuels available and their cost.
(5) Water supply.

Where the only available fuel is coal, coke or wood, there is at present no alternative to the steam engine, although the efforts being made to produce a "coal engine" will probably some day result in success—that is, an engine fed with coal directly and without the intervention of any separate boiler or producer.

When liquid fuel is available it may be used for firing the boiler of a steam vehicle, or it may, if of the right quality, be used in an internal combustion engine. Liquid fuels used at the present time in the latter kind of engine are either petrol, paraffin and benzol. Heavy residual oils may be used some day, but at present they are rarely employed, since the engines are exceedingly heavy.

At the present time most steam-driven vehicles use coal and most internal combustion-engined vehicles use petrol, although benzol and paraffin are occasionally used as substitutes for petrol. Probably this situation would continue were the price of petrol not too high; but the high cost of this fuel has impelled owners of motor vehicles to seek alternatives. Arrangements could be made largely to increase the amount of benzol available for this purpose, and continual efforts are being made to improve the apparatus for using paraffin. There is also the possibility that the supplies of petrol may be increased by some means of extracting petrol chemically out of paraffin. These various possibilities make it diffi-

cult to tell what is likely to be the most used fuel in the future. The problem is complicated by the possible success of the coal engine. Coal is very largely used in gas producer plant in conjunction with gas engines, and although such combinations are by reason of their weight unsuitable for road vehicles, there is a possibility that a simplified form of producer might be made little more bulky than a paraffin vaporiser, and so admit of use on the engine of an automobile. It would be a very great saving in cost if the cheapness of coal could be combined with the high economy of the internal combustion engine. A quantity of coal containing as much energy as a gallon of petrol can be purchased for about one penny.

If this should prove feasible it is doubtful whether even the superior thermal economy of the Diesel engine would enable it to replace the ordinary type. The Diesel engine has achieved a higher degree of thermal efficiency than any other form of internal combustion engine, because of its high compression and its improved "power factor." In place of a compression pressure of 90 lb. per sq. inch as with petrol, or 65 lb. per sq. inch as with paraffin, in the ordinary type, we here have a compression running up to 500 lb. per sq. inch. And instead of a maximum pressure very greatly exceeding the average, we have much more uniformity. The following table shows the relative efficiencies on the "air standard" basis :—

	Compression Pressure (Gauge).	Corresponding Compression ratio (approximate).	"Air Standard" Efficiency.
Ordinary cycle. Petrol .	90	4	0·43
„ „ Paraffin	65	3	0·36
Diesel engine. Crude oil	500	12	0·63

This table shows that the Diesel has a theoretical advantage of about 50 per cent. over either rival.

LOADS TO BE CARRIED—For passenger traffic the internal combustion engine is almost always preferred to the steam engine, but when heavy loads are to be carried the steam vehicle is sometimes used particularly for loads exceeding 5 tons. Such heavy loads may be carried on lorries or on trailers drawn by tractors. For loads ranging from 6 to 10 tons or even more it is almost always the practice to use steam tractors and trailers. The internal combustion engine tractor has only been developed in recent years and comparatively few are in use. Doubtless, however, their use will grow. Attempts, it is true, have been made on some scale to introduce road trains propelled by petrol engine tractors, but their use is far from common.

GRADIENTS—The effect of gradient is very important. If loads have to be taken by a hilly route more work has to be done by the engine. Climbing a gradient may easily quadruple the total resistance, causing a substantial rise in the H.P. needed if the speed is to be kept up to a reasonable level. The energy used in doing work against gravity is returned, it is true, on the descent of the gradient, but most of it is lost in the grinding action of the brakes and the engine is the loser on the balance of the account.

FUEL AND WATER SUPPLY—The consumption of water in a petrol engine is practically nil. A steam wagon, on the other hand, will certainly need a fresh supply every 20 miles, if not oftener. As regards fuel, the steam vehicle is better off and can commonly be made to run twice this distance with one supply of coal; but the petrol vehicle can easily beat the

steam vehicle in this respect, since enough fuel for 100 miles or more can be carried. For a sparsely populated or semi-developed country therefore, the petrol vehicle has great advantages, though it may be somewhat more expensive in first cost.

CHAPTER IV

Relationship of engine dimensions and gear ratios to work to be done—Design of vehicles propelled by (1) internal combustion engines, (2) steam engines—Wheel diameters—Braking

In the course of the last few years the manufacture of motor vehicles has grown to be one of the largest and most important of English industries. But little has been published which would enable the size of an engine and the appropriate gear ratios to be deduced directly from the conditions which a given motor vehicle has to fulfil. To some extent, this may be due to the absence of the test data necessary to enable standard relationships to be established. There have been innumerable bench tests of engines, enabling the value of the brake mean effective pressure (ηP) to be deduced for various sizes, shapes and speeds of engine, but until the Royal Automobile Club in July, 1912, carried out an organised series of tests[1] at Brooklands, there were scarcely any published results of the B.H.P. of complete cars. The report on these particular tests showed that the value of the brake mean effective pressure when the car is running at full speed on the level is little more than two-thirds of its value at the "hump" of the torque curve, the reason being that as the engine is then running about twice as fast, the air and petrol vapour do not have time completely to fill the cylinder by the end of the suction stroke. This factor, as will be seen, is an important

[1] See Appendix II.

one in car design. The problem of finding a fundamental basis for design is attractive, and in what follows a method of solution is given.

BRAKE MEAN PRESSURE—The term ηP is used in its usual significance, viz., that P is the mean effective pressure in lb. per square inch, as determined, say, from indicator tests, and η the ratio of B.H.P. to I.H.P. It has been ascertained by repeated tests upon petrol engines that the value of ηP is practically independent of the bore and stroke of the engine. That it should be independent of the stroke is at first sight surprising, since in steam engine practice to lengthen stroke is usually to lower the mean pressure. In the internal combustion engine, an increase of stroke is not accompanied by a higher compression ratio, since the latter must, in any case, be kept within the pre-ignition limit; identity of compression ratio means identity of compression pressure; then, since the richness of the mixture coming from the carburetter is not altered, we have the same rise of pressure due to explosion; this means the same maximum pressure, and, as the expansion ratio has not been altered, the same mean pressure.

One consequence of this independence of bore or stroke is that the average torque is directly proportional to total cylinder volume. Thus, if an engine be put to run against a fan brake, all engines of the same cylinder capacity would rotate the fan at the same speed, and therefore all would be running at the same H.P. This suggests that H.P. as well as torque is proportional to cylinder capacity, which, however, is not the case except when the conditions are such—which they usually are not—that the engines run at identical speeds.

ηP AND TORQUE—It is desired to obtain a relationship between mean pressure and torque. Let ηP be the mean pressure in lb. per square inch, and T the torque in

inch-lb., whilst d and l are cylinder bore and stroke in inches; then—

$$\text{Total pressure on piston} = \frac{\pi}{4}d^2 \times \eta\text{P lb.}$$

Work done per explosion—

$$= \frac{\pi}{4}\,d^2 \times \eta\text{P} \times \frac{l}{12}\text{ ft. lb.}$$

Taking the engine as four-stroke, as usual, we have, work done per cycle of two revolutions—

$$= \frac{\pi d^2 \eta \text{P} l}{48}\text{ ft. lb.}$$

Now this must be equal to the work done in two revolutions by the equivalent torque.

Therefore—

$$\frac{\pi d^2 \eta \text{P} l}{48} = \frac{\text{T}}{12} \times 2 \times 2\,\pi$$

so that—

$$\text{T} = \frac{d^2 \eta \text{P} l}{16}.$$

Now, $\frac{\pi}{4}d^2 l$, the cylinder volume, may be written C, in which case we have—

$$\text{T} = \frac{\eta\text{P.C}}{4\pi} \quad . \quad . \quad . \quad . \quad (1)$$

If, then, a motor vehicle be fitted with an engine of cylinder volume equal to C cubic inches—and it obviously does not matter for this purpose whether the volume be distributed in two or more cylinders or be concentrated in one—we know that a torque equal to that given by formula (1) can be obtained. To get the value in figures, we should require to know the value of ηP. Now, experiment shows that ηP depends very much on the engine speed. The diagram in Fig. 19 illustrates this. The value of ηP will usually be at its maximum—say, 95 lb. per square inch—at an engine speed of about

800 r.p.m. At higher speeds the gaseous mixture has not time to enter and fill the cylinder, with the consequence that ηP falls in value. It also drops somewhat at very low speeds, on account of the greater time there is for the expanding gases to cool or from incorrect timing of the ignition. From this it will be seen that for the propulsion of a motor vehicle it is not enough merely to settle the amount of engine torque that is needed, but attention has in addition to be given to see that the gear ratios from engine to road wheels are such as to enable the engine to run at the specific speed at which the desired torque is obtainable. Thus, if in the diagram—Fig. 19—an amount of torque equivalent to $\eta P = 75$ were needed at a certain car speed, and the gear ratios were such that at this speed the engine would have to work at 2,000 r.p.m., it is clear that the engine would decline to take up the load.

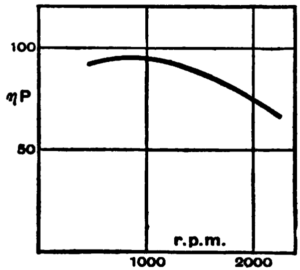

FIG. 19.—Typical Curve of ηP.

TRACTIVE EFFORT AND GEAR RATIOS—Manifestly, therefore, it is necessary to establish a relationship into which the gear ratios enter. Let G = the gear ratio, *i.e.*, the number of revolutions made by the engine for one revolution of the rear road wheels; let W = weight of vehicle in tons; and D = diameter of rear road wheels in inches.

Then the engine torque T when transmitted to the rear road wheels—neglecting frictional losses in transmission for the moment—will be $G \times T$, and the tangential force corresponding to this will be

$$GT \div \frac{D}{2} \text{ lb.}$$

$$= \frac{2\,GT}{D} \text{ lb.}$$

This may be expressed as lb. per ton of tractive effort by dividing by W, thus

$$\text{Tractive effort} = \frac{2\ \text{GT}}{\text{WD}}\ \text{lb. per ton} \quad . \quad . \quad (2)$$

We may now replace T from equation (1), and obtain

$$\text{Tractive effort} = \frac{\eta \text{P}}{2\pi} \cdot \frac{\text{GC}}{\text{WD}}\ \text{lb. per ton} \quad . \quad (3)$$

The tractive effort required depends on the work the vehicle has to do. If it has to climb a slope of 1 in 10, the tractive effort will need to be one-tenth of a ton per ton, or 224 lb. per ton. If it has to climb 1 in 4, the tractive effort will need to be one-quarter of a ton per ton, or 560 lb. per ton. Alternatively, it may be desired to achieve a high acceleration. A tractive effort of, say, one-tenth of a ton per ton would give an acceleration of one-tenth that of gravity, or an acceleration of 3·2 feet per second per second. The general relation enabling the tractive effort in lb. per ton needed to give an acceleration of so many feet per second per second to be computed is to multiply the acceleration by the constant 70 (*i.e.*, 2240 ÷ *g*). Thus, an acceleration of 3·2 feet per second per second would be produced by a tractive effort of 3·2 × 70 = 224 lb. per ton. A further consideration that needs to be taken account of is that the vehicle may be required to travel on very heavy roads, or, alternatively, at speeds so high that the resistance due to the air becomes very considerable. Thus, a typical touring car may be expected to encounter the resistances given in the following table, where V is speed in miles per hour, and R is the resistance in lb. per ton (measured at the clutch) :—

V :—10, 20, 30, 40, 50, 60.

R :—56, 74, 104, 146, 200, 266.

If this car were required to be capable of climbing on top gear (direct drive) a gradient of 1 in 100 at a speed

of thirty miles per hour and to accelerate, when desired, to the extent of, say, 0·65 feet per second per second, the tractive effort (measured at the clutch) necessary would be computed as follows :—

	Lb. per ton.
Necessary to overcome road and air resistance at 30 miles per hour	104
Necessary for climbing a slope of 1 in 100 .	22·4
Necessary for acceleration of 0·65 ft. per sec. per sec. = 0·65 × 70	45·5
Total	172 lb. per ton.

Having obtained this figure, it is possible to choose values of G, C and D, in equation (8), which will enable the necessary tractive effort to be obtained. To take a concrete case, we will assume that it is convenient to have D = 34 inches, W = $1\frac{1}{2}$ tons, and that G = 8·5 gives as high an engine speed as is desirable from the point of view of vibration and wear and tear when the car is running at top speed. Further, we will put ηP at the conservative figure of 70 lb. per square inch. Then we have—

$$172 = \frac{70}{2\pi} \times \frac{8 \cdot 5}{1 \cdot 5 \times 34} \times C$$

$$\text{or} \quad C = \frac{172 \times 2\pi \times 1 \cdot 5 \times 34}{70 \times 8 \cdot 5} = 225 \text{ cubic inches,}$$

which is the size of a four-cylinder engine of 4-inch bore and 5-inch stroke, and of 25·6 H.P. by R.A.C. rating. An engine of this capacity could do the specified work.

VALUES OF ηP.—The correct value to assign to ηP is a matter requiring a little discussion. As we have said, the maximum value, *i.e.*, height AC in Fig. 20, may be put at 95, being the highest normal value ηP has at the most favourable engine speed. When the vehicle is climbing hills the engine speed always falls back automatically to its best climbing speed—and if it cannot then surmount the hill the engine will stop unless the gear be

changed. Thus, if AB be the curve of ηP and EB be the curve of tractive resistance to motion shown in the same units, there will, at the point A, be a surplus available equal to AE, and this measures the hill-climbing ability. At B there is equilibrium between effort and resistance, and this point, therefore, is the running point when the car is at full speed on the level. The height BD is, of course, less than AC, and may, in general, be put as about equal to 67 lb. per square inch, viz., the value (67·2) assumed by the Royal Automobile Club Committee when formulating the R.A.C. rating. In using formula (3), therefore, it is advisable to put ηP = 67 when calculating the tractive effort corresponding to the car being at top speed on the level. The formula then becomes—

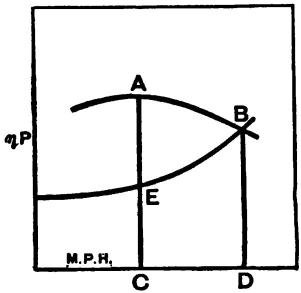

FIG. 20.—ηP and Tractive Resistance Curves plotted to same scale.

$$\text{Tractive effort} = 10{\cdot}7\,\frac{GC}{WD}\ \text{lb. per ton} \quad . \quad . \quad (4)$$

(or tractive effort $= 16{\cdot}6\,\frac{GC}{WD}$ lb. per ton, where C is in cubic centimetres and D in millimetres). If the car, running at top speed on the level, were to meet a gradual slope, the speed would fall off and the value of ηP would increase gradually from 67 to a maximum of about 95. If the hill continued to increase in steepness, the engine would stop unless the gear were changed. The effect of gear changing is to increase the value of G in equation (3) so that more tractive effort is produced, but the effective value of ηP is decreased through the loss of power in the gear box or back axle when the drive

is no longer direct. There is also an increased hysteresis loss in the driving tires at high values of tractive effort. These losses may be looked upon as a back pressure of, say, 10 or 25 lb. per square inch as the case may be. In what follows, the loss will be taken as 22 lb. per square inch on bottom gear and half as much on intermediate gears. (There is, however, some evidence that the loss on intermediate gears may be more than half.) This would make the efficiency (at maximum torque) from crankshaft to road $78 \div 95$ or 77 per cent. on bottom gear and $84 \div 95$ or 88 per cent. on the intermediate gear. Prof. Riedler[1] found the efficiency from clutch to road wheels on a car having all indirect gears to be from 75 to 84 per cent., but his tests on cars having direct drive on top are not yet published. From accelerometer tests on hill climbs, however, it seems that for cars having a direct drive on top there is a drop of efficiency of about 10 per cent. between a gear box ratio of 2, and one of 4.

As an example we will take the case of the engine studied above, which had a cubic capacity of 225 cubic inches. Let the gear ratio on bottom gear be $G = 14{\cdot}0$. Effective $\eta P = 95 - 22 = 78$.

Then

$$\text{Tractive effort} = \frac{78}{2\pi} \times \frac{14{\cdot}0 \times 225}{1{\cdot}5 \times 84}$$

$$= 720 \text{ lb. per ton,}$$

which means that when climbing a good hard slope having a resistance not exceeding 50 lb. per ton, there would be 670 lbs. per ton available for hill climbing, equal to a gradient of 1 in $8\frac{1}{3}$. Now, it is very unlikely that a hill so steep as this will be encountered; 1 in 5 is as much as average cars can climb steadily, and there are few countries where this hill-climbing power is not

[1] Wissenchaftliche, Automobil-Wertung, Berichte, VI.-X., 1912. (Berlin Technical School).

enough. This car when on such a gradient would have an excess tractive effort of (670 — 448) = 222 lb. per ton, which would produce an acceleration up the hill of 222 ÷ 70 = 3·2 ft. per second per second.

COMPLETE DESIGN OF MOTOR VEHICLE FROM PRESCRIBED CONDITIONS—The following are taken as typical requirements:—A motor wagon is to carry a load of 2 tons. It must be capable of travelling at 18 miles per hour on a good level road. It must be capable of climbing a gradient of 1 in 12 on the gear next to top gear, and must be capable of ascending 1 in 7 on bottom gear, the surfaces being good in each case. The body is to be a tilt one of a form for which the air resistance has been determined by previous tests, and we will assume that the tractive resistance measured at the clutch is given by $R = 45 + 11\left(\frac{V}{10}\right)^2$ lb. per ton. It is desired that if possible the vehicle should be capable of climbing steadily on top gear a slope of 1 in 30 without changing gear, but this is not considered essential provided it can nearly do so. Further, it is preferred that 34-inch rear road wheels should be used if possible.

SOLUTION—Since the vehicle is to carry a 2-ton load the total laden weight of the vehicle will probably be $4\frac{1}{4}$ tons. W, therefore, equals $4\frac{1}{4}$; D will be taken as 34 ins., as a trial.

Since $R = 45 + 11\left(\frac{V}{10}\right)^2$, the resistance at full speed—18 miles per hour—will be $R = 45 + 36 = 81$ lb. per ton.

Using now equation (4)—

$$81 = 10{\cdot}7\,\frac{GC}{4{\cdot}25 \times 34}$$

or

$$GC = \frac{81 \times 4{\cdot}25 \times 34}{10{\cdot}7} = 1{,}090.$$

This gives alternative values for G and C as shown in this table:—

G,	10,	9,	8,	7,	6,	5.
C,	109,	121,	136,	156,	182,	218.

Any pair of these may be taken, but, on the other hand, it is best that the engine speed should not exceed 1,400 r.p.m. at full speed on the level. If $G = 10$ were chosen, the engine would require to run about 1,800 revolutions per minute,[1] which is too fast. $G = 7{\cdot}5$ will be a better rate, and then $C = 145$ cubic inches. We will assume with the best modern practice that there are four cylinders, and that the stroke-bore ratio is 1·4. We then have $\frac{\pi}{4} d^2 \times 1{\cdot}4\, d \times 4 = 145$, from which we have $d = 3\frac{1}{4}$ ins. nearly. Our engine is, then, a four-cylinder one, $3\frac{1}{4}$ ins. $\times$ $4\frac{1}{2}$ ins. Taking these exact figures, the value of $C = 149$. On the gear below the top gear it is desired to climb a slope of 1 in 12, corresponding to a tractive effort of 187 lb. per ton; to this add the probable value of R—say, about 50—so that the total tractive effort must be 237. Applying equation (3) and putting the effective value of $\eta P = 84$, we have—

$$237 = \frac{84}{2\pi} \times \frac{149 \times G}{4{\cdot}25 \times 34}, \text{ or } G = 17{\cdot}2.$$

This gives the required gear ratio.

On bottom speed we require a tractive effort of $\frac{2,240}{7} =$ 320 lb. per ton, plus the probable resistance of 45 lb. per ton, making a total of 365 lb. per ton. Putting the effective value of ηP on this gear $= 73$, we have $G = 30{\cdot}8$.

Having obtained this skeleton design we can draw the three torque curves as in Fig. 21, where equivalent tractive effort is plotted vertically, and speed in miles

[1] $G = \frac{ND}{336V}$.

per hour horizontally. The three curves are very equally placed, and they form a homogeneous design. The best way to see whether the wagon could climb 1 in 80 on top gear is to raise the resistance curve parallel to itself through a vertical distance of 75 lb. per ton, as shown dotted. Since this curve and the torque curve on top gear fail to cross one another, it is then apparent that this

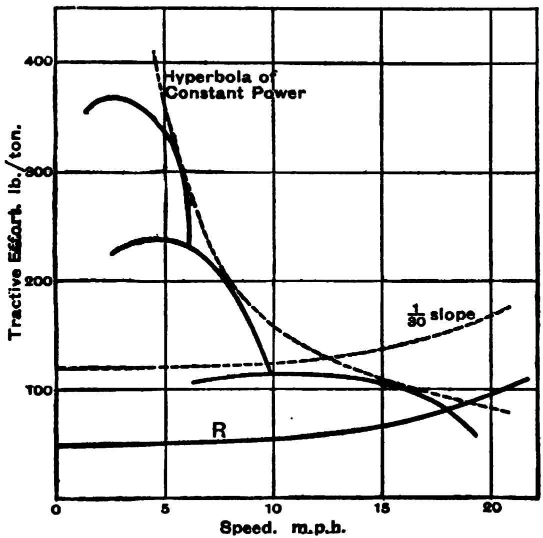

FIG. 21.—Design Chart for Motor Wagon.

hill is too steep to be climbed on this gear, the steepest climbable being 1 in 87. It remains to be seen whether this general design would be easy to drive. The test for this is that the three torque curves should approximate as closely as may be to the rectangular hyperbola—shown dotted—which represents the torque curve for an infinitely variable transmission gear worked by the same engine. The curve sheet is satisfactory in this respect. It will be noticed that a certain dropping away of torque with speed

tends to assist this approximation; nearly straight torque curves would make the vehicle difficult to drive. So that the artificial restriction of valve lift, or of carburetter feed pipe diameter, usual to prevent the engine being unduly raced, is also of use in another way. The curve sheet shows many interesting things. Thus, at 8 m.p.h. the wagon could steadily climb on top gear a slope of 1 in 87 [1]; or if on the level road could accelerate at 0·87 feet per second per second; or on second gear could climb 1 in 17, or accelerate on the level at 1·85 feet per second per second; it can also be predicted that if, when steadily climbing a slope of 1 in 25 under reduced throttle on second speed, the throttle were suddenly opened, the wagon would accelerate at 0·58 feet per second per second.

It will be found useful as an exercise in design to take the conditions prescribed by the War Office for their subsidy vehicles and deduce from them the most suitable dimensions for the vehicle to produce the best all round design.

TOURING CARS—As a corresponding example, the case of a touring car may be briefly dealt with. A typical instance is one in which the car is to be of the four-seater, open, touring type, weighing about 28 cwt. when fully laden, and just capable of climbing steadily 1 in 20, 1 in 10 and 1 in 5 on its three gears. It is to be capable of about 40 m.p.h. on the level, with an engine speed not exceeding 1,800 r.p.m. and its resistance formula is known from previous experience to approximate to $R = 50 + 6\left(\frac{V}{10}\right)^2$. Here we have enough data to enable the tractive effort on each gear to be calculated.

On lowest gear the tractive effort needed to climb

[1] A sliding cardboard chart to show this kind of action more clearly is sold at 1*s*. by Messrs. Elliott Brothers, of Central Buildings, Westminster.

1 in 5 is 448 lb. per ton. Add 52 lb. per ton for R at the probable climbing speed, and the total tractive effort equals 500 lb. per ton. On second gear the corresponding figure is 290 lb. per ton, and on top gear 185 lb. per ton. These are all hill-climbing figures. From these figures we can calculate the values of GC as in the previous instance. Thus, on top gear—

$$185 = \frac{95}{2\pi} \times \frac{GC}{1{\cdot}4 \times 32}, \text{ taking } D = 32 \text{ as a trial.}$$

Then

$$GC = \frac{185 \times 2\pi \times 1{\cdot}4 \times 32}{95} = 545.$$

Now, if the car speed is to be about 40 m.p.h. when the engine revolutions do not exceed 1,800 r.p.m., the value of G must not exceed 4·25. We will put it at this figure to begin with. Then $C = \frac{545}{4{\cdot}25} = 128$ cubic inches, equivalent with a stroke bore ratio of 1·50 to a four-cylinder engine of about 3 inches × 4½ inches, or 76 mm. × 115 mm. Using equation (3), we also obtain the other gear ratios, giving the series, 4·25, 7·5 and 15·0. This gives a rough outline of the car which further refinements can complete.

Racing Vehicles—One of the most remarkable performances by a racing car was that of a 25·6-H.P. Talbot car which covered over 100 miles in one hour on the Brooklands track. In the *Autocar*[1] there have been given some particulars of this car which throw a good deal of light on its design. It is stated that the B.H.P. follows a perfectly straight line law from 20 H.P. at 500 r.p.m. to 120 H.P. at 3,000 r.p.m.; after 3,000 r.p.m. the H.P. rises less rapidly, until at 3,750 r.p.m. 133 H.P. is obtained. The maximum speed of this car was 105 m.p.h. equivalent, with 880 mm. wheels and a

[1] February 22nd, 1913.

gear ratio of 2·48, to 2,650 r.p.m., at which speed the bench B.H.P. was 118. The crankshaft torque corresponding to this power is

$$T = \frac{118 \times 33{,}000}{2{,}650 \times 2\pi} \times 12 \text{ inch-lb.}$$

$$= 2{,}700 \text{ inch-lb.}$$

The engine was a four cylinder one of 101·5 × 140 mm. bore stroke, so that the cylinder capacity would be 4·53 litres or 276 cubic inches.

Whence it follows from equation (1) that

$$\eta P = \frac{4\pi T}{C} = \frac{4\pi \times 2{,}700}{276}$$

$$= 122 \text{ lb. per square inch,}$$

which is very far above the value of ηP which we have found usual even at much lower engine speeds with the non-racing engines fitted to touring cars and motor wagons.

The tractive force may be calculated from

$$\text{B.H.P.} = \frac{R \times W \times V}{375},$$

therefore $RW = \frac{375}{V}(\text{B.H.P.}) = \frac{375}{105} \times 113 = 400$ lb. approximately. If, therefore, the weight of the car were in the neighbourhood of 1 ton, the value of R would be 400 lb. per ton, corresponding roughly to a tractive resistance formula of the form—

$$R = 50 + 3\left(\frac{V}{10}\right)^2,$$

showing that the coefficient of the V^2 term was only about half the amount it would be in an ordinary touring car. Reduction of windage area and the use of the minimum quantity of gear box lubricant were the probable means of this alteration.

SUMMARY—The three chief relationships in the design of motor vehicles are the following:—

(1) $$T = \frac{\eta P \cdot C}{4\pi}$$

(2) $$\text{Tractive effort} = \frac{2\,GT}{WD} \text{ lb. per ton.}$$

(3) $$\text{Tractive effort} = \frac{\eta P}{2\pi} \cdot \frac{GC}{WD} \text{ lb. per ton.}$$

The third of these may be put in the form—

$$\text{Tractive effort} = 16{,}600\,\frac{GC}{WD},$$

where ηP has been written at its R.A.C. rating figure, and where C is expressed in litres, and D in millimetres. This is sometimes convenient.

It is not claimed that the methods of design given above are incapable of improvement. Most of the factors are susceptible of adjustments which may bring the designs better into line with other models or may reduce the cost of manufacture. But by following the theory above given it is quite easy to study the possible permutations and combinations, and eventually to find a solution embodying the maximum of good points with the minimum of bad ones. And that is as much as can be expected.

THE "GEAR" OF A MOTOR VEHICLE—The gear ratios by themselves tell one little unless the diameter of the rear road wheels be also stated. It would be more convenient to have some property combining the two. This we may obtain by dividing the wheel diameter by the gear ratio and calling the result the "gear." Its dimensions are obviously linear, and it represents the diameter of a wheel on the crankshaft which would have a peripheral speed equal to that of the vehicle itself. Thus, with 34-inch wheels and a gear ratio of ten, the "gear" would be 3·4

inches or 87 mm. This use of the word "gear" is precisely the same as that introduced in connection with push-cycling many years ago.

In the case of the touring car recently described, in which gear ratios of 4·25, 7·5, and 15·0 were found with a road wheel diameter of 32 inches, the "gears" would be said to be 7·53, 4·27 and 2·13 inches.

This method of using the idea of "gear" becomes useful when comparing motor vehicles with one another. Thus, if we denote the "gear" by the letter γ we may write equation (8).

$$\text{Tractive effort} = \frac{\eta P}{2\pi} \cdot \frac{GC}{WD} = \frac{\eta P}{2\pi}\,\frac{C}{W\gamma}\ \text{lb. per ton.}$$

This shows that the tractive effort for a given ηP is proportional to the ratio of the cylinder volume per ton to the "gear." And since maximum H.P. in hill climbing is proportional to $\eta P \times C$, we get the relationship (which is easily seen to be correct)

$$\text{Tractive effort in lb. per ton} \propto \frac{\text{H.P. per ton}}{\text{Gear}}.$$

It is of interest to apply the formula

$$\text{Tractive effort} = \frac{\eta P}{2\pi} \cdot \frac{C}{W\gamma}\ \text{lb. per ton}$$

to the case of a number of standard touring cars.

The table given below is obtained by selecting from the pages of the *Automobile Engineer Yearbook* for 1912, the first one, two or three cars in each of the capacity groups and estimating their weight on the road by adding 75 per cent. to the chassis weights given in the *Autocar* list. The value of ηP is the R.A.C. figure of 67 and roughly corresponds to its real value when running all out on the level on top gear. This value of ηP needs to be increased for hill-climbing conditions by from 5 to 50 per cent. depending on the gear in use. But by

taking $\eta P = 67$ throughout it is possible to get a rough comparison of car with car.

Car No.	Capacity, litres.	Probable weight laden, tons.	$\frac{D}{G} = \gamma$ min.				$\frac{C}{W}$.	Tractive effort at clutch. Lb. per ton.			
			1st.	2nd.	3rd.	4th.		1st.	2nd.	3rd.	4th.
	C.	W.									
45	2·03	1·22	44	100	125	180	1·67	630	280	220	155
47	2·12	1·10	60	108	190	—	1·92	530	295	165	—
120	3·01	1·22	82	136	203	272	2·46	500	300	200	150
121	3·01	1·42	68	104	153	226	2·12	520	340	230	155
177	4·07	1·49	67	132	276	—	2·72	670	340	165	—
178	4·07	1·79	56	98	131	242	2·26	670	380	285	155
179	4·08	1·52	64	122	191	273	2·68	690	365	230	160
225	5·15	1·84	76	112	175	271	2·80	610	415	265	170
Average tractive effort figures for the six 4-speed cars . .								603	348	238	158

These figures indicate that the tractive effort on a 4-speed car is 600, 350, 240 and 160—in round figures—on each gear. All this assumes $\eta P = 67$. For bottom gear ηP will not greatly exceed this figure—probably not by more than enough to allow for the probable road resistance—so that the maximum grade climbable would be 600 ÷ 2,240, or about 1 in $3\frac{3}{4}$. On top gear the hill-climbing value of ηP would be about 50 per cent. higher, so that a tractive effort of 240 lb. per ton might be attained, giving a hill-climbing capacity of 1 in 15 with 90 lb. per ton tractive resistance at the clutch, and a correspondingly less gradient for higher resistance figures.

STEAM VEHICLES—We saw (equation 3) that in the case of internal combustion engines

$$\text{Tractive effort} = \frac{\eta P}{2\pi} \cdot \frac{GC}{WD}.$$

Now this assumes a four-stroke cycle and a single acting engine. Steam engines, however, are two-stroke and double acting, so that their tractive effort is four times that given by the above formula, provided that P be still kept to represent the mean pressure of any working stroke. For the steam engine, therefore, we have

$$\text{Tractive effort} = \frac{2\,\eta P}{\pi} \cdot \frac{GC}{WD} \quad . \quad . \quad (5)$$

When compound expansion is used, the product PC is made up of two parts, that due to the high pressure cylinder and that due to the low pressure cylinder. We have, therefore, $PC = P_1C_1 + P_2C_2$ where the high pressure cylinder is denoted by the suffix 1 and the low pressure by the suffix 2.

Now P has not the almost constant value in steam engines which it has in petrol engines. It depends upon the boiler pressure and upon the setting of the slide valve. For a given boiler pressure there is, however, one setting that gives the highest value to P, and we may write

$$P = k \,.\, B,$$

where B is the boiler pressure and k is a constant.

Substituting this we have

$$\text{Tractive effort} = \frac{2\,\eta k B}{\pi} \cdot \frac{GC}{WD} \quad . \quad . \quad (6)$$

And this is the most convenient equation to use.

The value of ηk will depend on the type of engine and its mode of action, but to a first approximation it may be put as $\frac{1}{3}$, giving the approximate formula

$$\text{Tractive effort} = \frac{2\,B}{3\pi} \cdot \frac{GC}{WD} \text{ lb. per ton.}$$

Example—If, for instance, the boiler pressure be 200 lb. per sq. inch (gauge pressure); the total cylinder capacity be 350 cubic inches (*i.e.*, a compound engine 4 inches by 7 inches and 7 inches by 7 inches); diameter of tractor wheels 48 inches; total weight laden = 12 tons, and if the gear ratios on top and bottom speed be 13 and 26 respectively, then we have

On top speed,

Tractive effort $= \dfrac{2 \times 200}{3\,\pi} \times \dfrac{13 \times 350}{12 \times 48} = 335$ lb. per ton, showing a hill-climbing ability with a tractive resistance of 80 lb. per ton equal to a grade of $\dfrac{335-80}{2{,}240}$ or 1 in 8·8.

On bottom speed,

Tractive effort $= \dfrac{26}{13} \times 335 = 670$ lb. per ton, showing a hill-climbing ability on a similar road of $\dfrac{670-80}{2{,}240}$ or 1 in 3·8.

This about fits the case of a 5-ton lorry capable of about 8 m.p.h. as a maximum, and giving about $1\frac{1}{2}$ G.T.M. per lb. of coal and 0·2 G.T.M. per lb. of water.

The figure $1\frac{1}{2}$ G.T.M. per lb. of coal can be used to get an overall efficiency figure if we assume some average calorific value for the coal. If, for instance, we take it at 11×10^6 ft. lb. per pound, and if the tractive resistance be 80 lb. per ton, then we have a resisting force of 80 lb. overcome for $1\frac{1}{2}$ miles for every 11×10^6 ft. lb. of fuel energy, so that

$$\text{Overall efficiency} = \frac{80 \times 1{\cdot}5 \times 5{,}280}{11{,}000{,}000} \times 100 \text{ per cent.}$$
$$= 5{\cdot}8 \text{ per cent.}$$

The loss is therefore 94·2 per cent., and includes heat lost by boiler cooling, by loss up the chimney, by engine

cooling loss, by exhaust loss, and by leakage. The efficiency is, of course, much below that of an internal combustion engine, but the lower initial cost is often regarded as a balancing factor.

When wood fuel is used instead of coal the weight of fuel used is about twice as great for the same work to be done. This limits the radius of action without replenishment of fuel to twenty miles or less. But if the vehicle be running in an undeveloped wooded country the necessary fuel is easily picked up by the way. The country needs also to be well watered for a steam vehicle to be used, since the customary supply carried will not suffice for more than, say, twenty-five miles.

WHEEL DIAMETERS—It is not possible in the design of heavy motor vehicles, petrol or steam, to pay attention only to engineering considerations. The rules of the Heavy Motor Car Order of 1904 have to be borne in mind. There it is provided that steel tires must never be less than five inches wide as a minimum; and that for wheels of three feet or *more* in diameter the tire width must at least be equal to

$$\text{axle load} \div (2\,D + 9),$$

where $D =$ wheel diameter in feet, and the axle load is measured in cwt.

For wheels of three feet or *less* in diameter the rule is

$$\text{minimum width} = \text{axle load} \div (4\,D + 8).$$

Moreover, with these tires the speed must not exceed 5 m.p.h. if there is a trailer, or if the tare weight exceeds 3 tons; but without a trailer and for a less tare weight than 3 tons a speed of 8 m.p.h. is permitted.

When, however, rubber tires are used on all four wheels, the speed may be 12 m.p.h. when the maximum axle load laden is not over 6 tons, and 8 m.p.h. when it is.

BRAKE DESIGN—Every motor vehicle is required by law to have two brakes. In practically all load carrying

vehicles (as distinct from load pulling vehicles, *i.e.*, tractors) these two brakes are the hand brake and the foot brake. The former is applied by hand and usually operates on brake drums attached to the rear road wheels, the transmission mechanism being such that the braking pull is equally shared by both wheels. The foot brake should properly also act directly on the rear road wheels, but in most designs it does not do so and works instead on the driving shaft coming away from the gear box; this arrangement has the great disadvantage that the braking force has to pass along the propeller shaft, through the bevel or worm gear and through the two cross shafts (and chains if there be any). There are very real drawbacks to this arrangement:—

(1) If any part of this transmission gear should fail under the stress the brake is at once rendered useless.

(2) The transmission gear may have to be made stronger than would be necessary were the engine drive only to be transmitted along it.

The foot brake is usually much the more powerful of the two brakes, and a retarding effort of as much as 700 lb. per ton can be reached. Now the engine drive on touring cars may reach this figure, but on motor wagons it is usually much less—rarely exceeding 400 lb. per ton. This shows that in the case of motor wagons the foot brake may cause the design of the transmission gear to be $\frac{7}{4}$ times as strong as it need be were the brake drum put in its proper place, *i.e.*, as close as possible to the work it has to do.

A limit to the braking power arises through the liability of the driving wheels to skid if too much braking be applied. If, for instance, the braking be 700 lb. per ton and if the total weight of the wagon be divided as $\frac{2}{3}$ on the rear axle and $\frac{1}{3}$ on the front, the braking effort

being all on the back axle is equal to 1,050 lb. per ton of axle load, or 0·47 of the axle load. This is not far from the point at which the wheels will skid—which usually occurs at from 0·4 to 0·6, depending on the tire and road surfaces.

A very rapid application of the brakes is undesirable, alike in the interests of passengers and goods. The retarding effort should rise gradually to a maximum and then be gradually released. The rate of increase or decrease of acceleration should not exceed a certain maximum, say, 6 ft. per sec^3.

WASTE OF ENERGY IN STOPPING—The energy used up by the brakes is lost beyond recovery, except in certain electric designs. This is why the descent of a hill does not compensate for its ascent, unless the gradients are very slight indeed. It also explains why it is so much more costly to run a service with frequent stoppages.

Suppose we have a given vehicle running at 20 m.p.h. Its kinetic energy per ton will be

$$\text{K.E. per ton} = \tfrac{1}{2} \cdot \frac{2{,}240}{32 \cdot 2} \cdot \left(\frac{20 \times 88}{60}\right)^2 \text{ ft. lb.}$$

$$= 30{,}000 \text{ ft. lb.} = 13 \cdot 4 \text{ ft. tons.}$$

And if there be the usual amount of rotational momentum we may add 5 per cent. to this, making a total of, say, 14 ft. tons per ton weight of vehicle. Now, if this vehicle be running on a service in which a stop every 200 yards is necessary, the space-average loss of energy due to this braking is

$$\frac{14 \times 2{,}240}{200 \times 3} \text{ ft. lb. per ft. per ton}$$

$$= 52 \text{ ft. lb. per ft. per ton}$$

$$= 52 \text{ lb. per ton.}$$

This shows that the average loss of energy due to the braking will nearly double the total energy consumption

per mile since the tractive resistance would probably not very greatly exceed 52 lb. per ton.

For a rapid service with frequent stops there is therefore good reason for using regenerative braking of some kind—such, for instance, as that on the Macfarlane-Burge system.

CHAPTER V

Electric and petrol-electric systems — Curve of ideal tractive effort—Hydraulic systems

ELECTRIC AND PETROL-ELECTRIC SYSTEMS—In a purely electric system the current is either drawn from a storage battery carried on the vehicle or from overhead conductors connected to some electric generating station. There are comparatively few storage vehicles, but many electric vehicles taking power from overhead. The latter, commonly called "trolley omnibuses," are growing in numbers rapidly.

In a petrol-electric system the electric mechanism is subsidiary and exists only because of the difficulty of inventing any satisfactory mechanical method of changing gear. As has already been stated, the ideal is one in which there would be an infinite number of gear graduations possible — so that the curve-connecting tractive effort and car speed would be hyperbolic. Electrically the problem is capable of satisfactory solution, and there are several ways in which it can be done. One is to drive a dynamo from the engine and to supply current to a motor or motors connected to the road wheels. A series motor has a characteristic very near to the ideal. Another is to drive both dynamo and car on the same shaft, storing the current in a battery when the car does not need all the torque the engine is producing, and drawing current from that battery when the engine is overloaded. Still another way is to drive mechanically from the engine to the road wheels, having an engine

large enough not to need help from any storage battery, and to use a combination of two electric machines, alternately generator and motor, as a fly-wheel and clutch. Many alternative arrangements are possible, since there is scarcely any limit to the ingenious manner in which electric power may be transformed and used.

It is not feasible to describe here all the various forms of electric and petrol-electric vehicle, and the types which are selected for mention are chosen either because they promise to be extensively used or because of their special ingenuity. A brief description is given of :—

(1) The Tilling-Stevens Petrol-Electric System.
(2) The Thomas Transmission, also petrol-electric.
(3) The Macfarlane-Burge Electric System.
(4) The Macfarlane-Burge Petrol-Electric System.

The descriptions are for the most part given in the words of those responsible for them.

(1) THE TILLING-STEVENS SYSTEM—This system has found very extended use, and is a petrol-electric type of the first class mentioned, viz., that in which there is a complete break between the crank shaft and the propeller shaft, so that all power is carried from one to the other electrically. The following account is that given by the firm :—

The "Tilling-Stevens" Petrol-Electric Vehicle is the outcome of many years' experience.

The elimination of the clutch and gear box and the substitution of a flexible electrical transmission between the engine and the driving wheels qualifies the vehicle for service in which easy starting and control, in addition to smooth, silent, speedy and economical running is desired.

The final drive is effected by an electric motor, driving by means of a cardan shaft, the worm gear on the back axle. The electrical supply for the motor is supplied by a specially constructed dynamo, direct coupled to the petrol engine, which is the mechanical source of power. This generating plant develops considerably more electrical energy than a battery four times its weight.

The electric control box has three positions, forward, neutral, and reverse, operated by means of a single lever. The dynamo field control is operated by means of a lever under the steering wheel. The driver controls the speed of the vehicle by means of a throttle pedal.

The electrical transmission consists of a generator driven directly by the engine by a spring coupling, a series wound electric motor direct coupled to the cardan shaft, and the controller box. The controller box carries a reversing switch, and a shunt resistance for the generator fields is carried in a separate box, both boxes being mounted on a dash and operated by two small levers. The whole electric transmission arrangement has been designed to ensure the greatest possible simplicity of construction and control.

The generator is capable of an output of from 1 to 20 kilowatts at a speed varying from 350 to 1,400 r.p.m. at a voltage varying from 0 to 300, and it has been carefully designed with a falling characteristic, so that any increase in the demand for current when the engine is fully loaded is accompanied by a corresponding reduction in voltage. The output in kilowatts at any speed is proportional to the power exerted by the engine, but the volts and amperes may vary over a large range according to the gradient, speed, or degree of acceleration required. The amperes required by the series wound motor are nearly proportional to the torque on the cardan shaft, and the speed of the motor is to a less degree proportional to the voltage of supply. It thus follows that when the vehicle is running on the level road the demand for amperes will be small, but on gradients it will considerably increase, with a corresponding decrease in voltage, resulting in a slower speed with increased torque; this change takes place automatically.

On level roads and ordinary gradients the whole of the control is effected by the throttle pedal operated by the right foot of the driver. On stiff gradients the shunt resistance has to be employed to allow of increased engine speed. The controller has three positions, forward, neutral, and reverse, the whole of the speed regulation being effected with the one position of controller. As the generator ceases to excite at 300 r.p.m. of the engine, it is not necessary to break the circuit between the dynamo and the motor on stopping in traffic, as by the release of the throttle pedal and the consequent slowing down of the engine no power is transmitted to the cardan shaft.

The field windings of the machines are so arranged that it is

impossible for the driver to "jerk" the vehicle even if the circuit between the generator and motor is closed when the former is being driven by the engine at a high speed, as the generator will not excite until the circuit to the motor is closed.

The electrical machines, which are of the semi-enclosed type, are ventilated by a disc fan carried on the end of the generator shaft remote from the engine, and revolving between the end covers of the generator and motor. The air inlets to the machines are carefully guarded, and easily removable covers are provided for access to the brush gear and commutators.

The overall commercial efficiency running in normal service has

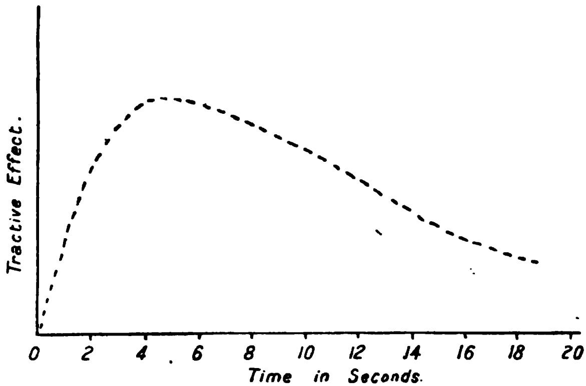

FIG. 22.—Tractive Effort Curves obtainable with Tilling-Stevens Transmission. (The maximum is at about 200 lb. per ton.)

been found to be 79 per cent., being 90 per cent. on the motor and 88 per cent. on the generator.

We claim great economy in upkeep owing to:

(*a*) The great simplicity of the transmission, no gear wheels, clutches or battery being employed.

(*b*) The control of the vehicle is the simplest yet employed, being on level roads and slight gradients entirely effected by the use of the throttle pedal, no circuits being broken in driving.

(*c*) Great economy in petrol consumption, owing to the engine speed averaging considerably less than that of the cardan shaft.

(*d*) Owing to the vehicle's great powers of free-wheeling the

engine is running with its throttle nearly closed a great part of the time.

(*e*) The elastic start up, with acceleration equal to a steam car, causing a minimum of transmission stress on the vehicle and tires, and the marked absence of vibration adding considerably to the life of the body.

(*f*) Increased life to the engine, running on an elastic electrical drive with no possibility of shocks from clutching and declutching, as is generally the case with vehicles having a mechanical drive alone.

The tractive-effort time curve for this system of transmission is as shown in Fig. 22, and it comes near the ideal. There is a gradual rise of acceleration instead of the sudden jerk which so often occurs in vehicles not having an electric transmission gear.

Its disadvantage is its low mechanical efficiency compared with a direct drive, and this is the more noticeable on light loads. Nevertheless, the overall efficiency—from fuel to road—is stated by the makers to compare favourably with that of other vehicles.

(2) THE THOMAS TRANSMISSION—This system has the advantage that when the vehicle has got up to speed the drive becomes mechanical, and there are then no electric losses to lower the efficiency. It has been very successful on test, and in 1911 was awarded the "Dewar Trophy" by the Technical and Expert Committee of the R. A. C. It is not a very easy system to understand, but the following account taken from the R. A. C. test certificate makes it fairly clear:—

Description of Transmission.—The system consists of three elements, (1) a planetary gear, (2) and (3) two electrical machines as shown in Fig. 23.

The casing (*a*) forms the fly-wheel of the engine and is fixed to the crank shaft. Inside the casing are two planet pinions (*b*) and (*c*), of different size but keyed to the same spindle, which is free to rotate in the sides of the casing. The planet pinions (*b*) and (*c*) mesh with the two sun pinions (*d*) and (*e*) which are respectively keyed to

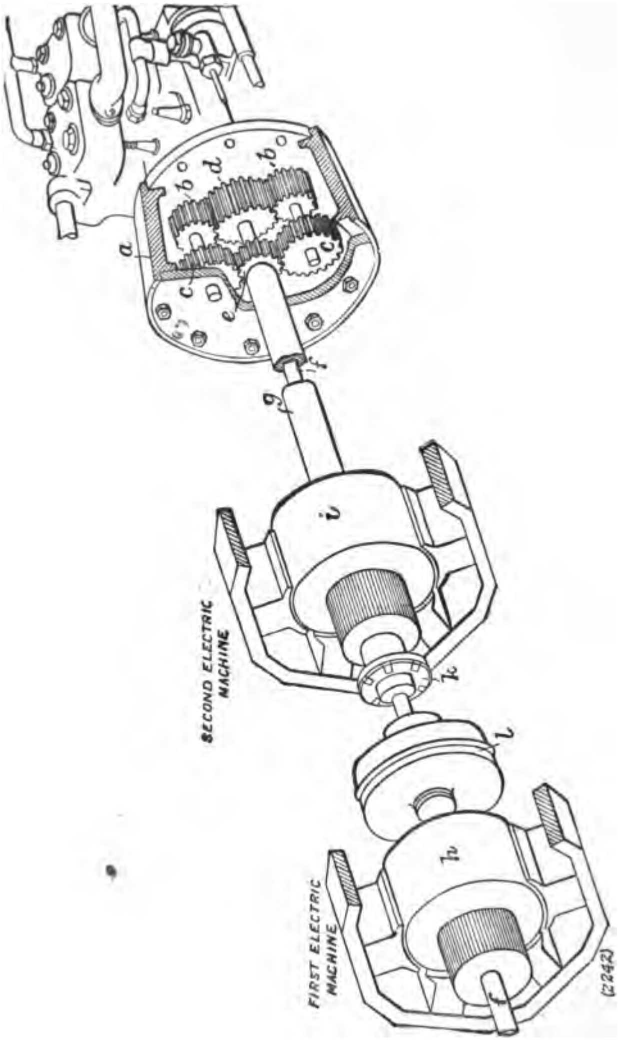

Fig. 23.—Diagrammatic view of the Thomas Transmission.

the independent shafts (f) and (g). On the shaft (g) is mounted the armature (i) of the one electrical machine, while on the shaft (f), which is connected to the propeller shaft of the car, is mounted the armature (h) of the other electrical machine. The two electrical machines are connected in series. There are thus two paths for the power transmitted by the engine to the road wheels; the first, a direct mechanical path from (d) along the shaft (f), and the second, an electrical path from (e) through the two electrical machines.

The pinion (d) is larger than the pinion (e); therefore, for a certain rotation of the casing (a), the pinion (e) (with the hollow shaft and the armature (i)) will tend to rotate backwards, while the pinion (d) (with the shaft (f) and the armature (h)) will tend to rotate forward, at speeds depending upon their relative resistances to motion. Thus, if the speed of (a) (*i.e.*, the engine speed) is constant, the speed of the shaft (f) will vary as the speed of the shaft (g), and since the speed of (g) depends upon its resistance to motion and, therefore, on the load of the second electrical machine, the speed of (f) (the propeller shaft) can be varied by the variation of power transmitted electrically between (i) and (h).

The operation of the transmission is as follows :—The casing (a) is rotated by the engine at approximately constant speed. Before starting there is no electrical connection between the first and second electrical machines, (e) (which is on the same shaft as the armature of the second electrical machine) rotating backwards, while (d) remains stationary. To start the vehicle current is taken from (i) to (h). This has a double effect upon the shaft (f). The current transmitted to (h) exerts a torque on (f), and the loading of (i) with this current, by decreasing the speed of (e), causes (d) to rotate. A part of the torque is thus transmitted electrically and a part mechanically. As (i) is gradually decreased in speed the vehicle increases in speed until (g) practically comes to rest. Up to this point the second electrical machine has been acting as dynamo and the first electrical machine as a motor.

Both machines now change their functions, power being transmitted from (h) to (i). Owing to (g) being forced to rotate against, and in the same direction as, the engine, the speed of (f) still further increases. The speed of (g) increases more rapidly than that of (f) (due to the gear ratio in the planetary gear) until, finally, they are both travelling at the same speed. The coupling (k) is then engaged and the current in the electrical circuit dies down to zero. The engine is now driving direct through to the

propeller shaft. The second electrical machine (connected up as a shunt machine) is used, when running on top speed, to charge a set of lighting cells, which are again used through the medium of the same machine to start the engine. The reverse is operated in the following manner. The coupling (*k*) is engaged and the clutch (*l*) disengaged. The armature (*i*) of the second machine is then rotated by the engine and the current transmitted to the first machine, which is given a reversed field.

The variation of the power transmitted electrically (and, therefore, of the speed of the vehicle) is obtained by varying the strength of the fields of the two electrical machines. This is done through the medium of a controller, making and breaking contact in oil. There are fifteen positions of the controller, including starting the engine—ten electro-mechanical "speeds," one direct top, one for charging the battery, and two reverse positions. All operations are controlled by the position of one lever.

In this system the braking is mechanical, not electrical. The starting acceleration is strikingly smooth and free from jerks. The system is being successfully applied to railway motor coaches.

The overall efficiency is very high, and the Thomas Transmission holds the "record" for gross ton mileage per gallon of petrol, for in an R.A.C. test of a Delahaye touring car fitted with this arrangement and driven from London to Edinburgh and back, no less than 35·73 miles were run per gallon, equal to 67·92 gross ton miles per gallon.

The claims made for this system by its supporters are that:—

(i.) There can be an unlimited number of gear ratios or speeds.

(ii.) The changing of speeds is effected very simply by the mere movement of a lever, and is unaccompanied by noise or shock; consequently there is less liability for the occurrence of breakages, and less general wear and tear.

(iii.) The transmission itself is very efficient over a wide range of vehicle speeds, and it permits of the engine being driven practically continuously at its most economical speed, thus resulting in a very low fuel consumption.

(iv.) Incidentally, since the electrical equipment, when on top speed, is not required for purposes of propulsion, it may be usefully employed in automatically keeping a small battery fully charged, so providing at all times a supply of current for lighting, and for starting the engine from the driver's seat. This battery is not essential to the transmission, but is a very useful adjunct.

(v.) The system is applicable to the operation of road trains. In this case the trailers are connected electrically to the tractor, and are assisted by the current at such times as a part of the power is being electrically transmitted.

(3) MACFARLANE-BURGE ELECTRIC SYSTEM — An account of this system was communicated to the Institution of Electrical Engineers in 1911–12. A storage battery which is carried on the vehicle supplies current to the driving motor through the medium of an "electric valve," which prevents the battery being damaged through careless driving. The "valve" acts in such a way as to diminish the volts supplied to the motor as the current rises, and thus automatically prevents the motor from causing any overdraft of current from the battery. The following extract from the I.E.E. paper above mentioned puts the matter in the inventor's words :—

The characteristics of this system are :—

(*a*) Braking entirely by regeneration, the battery absorbing the energy returned.

(*b*) A controller in the shape of a very efficient rotary transformer or automatic electric valve (only transforming half the power applied to the wheel motors) which

automatically limits the current that can be drawn from or returned to the battery, displacing the usual series-parallel type of controller.

(*c*) A motor with special shunt field windings, having a torque speed characteristic similar to a series motor.

The battery current is limited to a predetermined value under any conditions of speed, *i.e.*, the automatic valve only allows a safe amount of power to pass from the battery to the driving motor under the worst conditions of driving, and thus the capacity of the battery is retained.

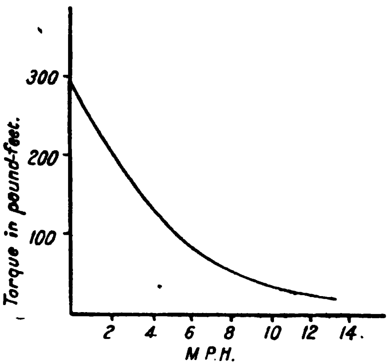

FIG. 24.—Torque-Speed Curve for Macfarlane-Burge Electric Gear.

Altogether it will be seen that the battery is used under very favourable conditions indeed, and it should also be noted that the regenerative action of the motor comes into play whenever the driver reduces the speed or stops the vehicle.

As traffic conditions compel a large number of starts and stops, the battery becomes more of the nature of a "buffer" battery, the life of which, as is well known, is greater than that of a battery being discharged during the whole time the vehicle is in commission. This has been demonstrated by applying a low-frequency alternating current to the terminals of a battery cell, discharging and charging the cell during 1 cycle, when it was found that the ampere-hour and watt-hour efficiency were extremely high, and the life very much greater than that under ordinary conditions.

The torque-speed curve is shown in Fig. 24, and as will be seen, it is an excellent one.

(4) MACFARLANE-BURGE PETROL-ELECTRIC SYSTEM.—A description of this system was also given in the I.E.E. paper already mentioned. In order not to draw continually on the storage battery, a petrol engine capable of

yielding the *average* power needed is used in conjunction with the electric system. The battery receives power when the engine power exceeds that needed for propulsion and yields it when the engine needs assistance. In this way only a small battery is needed, and the engine may be a good deal smaller than that necessary in purely petrol vehicles put to do the same work.

The following is an extract from the inventor's account of the system :—

Engine—The engine runs practically at constant speed, and has only to supply the average power required to drive the vehicle. It is well known that to drive a 6-ton loaded vehicle on the level at 12 miles per hour, 9 B.H.P. is required. Now, when running at about half-speed uphill, in this system the gear ratio automatically changes, enabling the full 9 H.P. to be utilised ; the torque, therefore, under this condition is twice that developed on the level. Further, with the aid of the battery, the motors are enabled to develop an additional 9 H.P., making a total of 18 H.P. at this speed, thus giving four times the normal running torque. A petrol vehicle, however, running up the same hill at half-speed on the top gear would have to be equipped with a 36-H.P. engine on the normal full-speed basis, because at half-speed it would only just be able to develop the 18 H.P. required. It may be argued that an earlier change of gear would enable a smaller engine to be used, but even this will not necessarily compensate for the falling off of power due to loss of compression and inefficient action of the carburetter, and it is a fact that 36 to 40-H.P. engines are actually fitted to such vehicles.

Battery—This is an ordinary buffer battery constantly charging or discharging automatically within safe limits, according to the nature of the road, and as such works under very favourable conditions. Besides acting as an equaliser, other important functions of the battery are :—

(*a*) To start up the engine from the driver's seat, a very great convenience, and to supply current to the sparking coils.

(*b*) To enable the vehicle to run to a garage in the event of an engine breakdown.

(c) To light the vehicle and to provide current for electric signs and advertisements.

(d) To allow of electric braking down to any speed.

STORAGE BATTERY WAGONS—Many attempts have been made to run motor vehicles with lead storage batteries, but in this country, at any rate, they have not been very successful. The vibration due to travelling over the road has been deleterious to them, and the spilling of the acid has been an ever-present difficulty. Moreover, the weight to be carried has been very great. Other reasons, too, have conspired to defeat the aims of those engineers who endeavoured to make a success of this method of traction. The average life of a lead storage cell[1] has been put at about 12,500 miles, and its capacity on a three-hour discharge as 3·5 watts per pound of total weight.

There are stated[2] to be as many as 20,000 electric road vehicles in the United States, of which some 6,000 are "trucks." If a hardy type of battery were available there would be a general increase in the number of vehicles used, especially as it is possible that central generating stations would be willing to supply current during light load hours at no more than ½*d.* per unit. Much interest attaches to the test made by the R.A.C. in 1913 of a motor wagon driven by Edison storage cells. The Edison cell has had much published about it, but the extensive claims sometimes made have created a certain amount of prejudice. The publication of the result of an R.A.C. test has more than ordinary interest, therefore, as affording evidence of the degree to which the claims made can be fulfilled in practice.

The Edison cell, as is well known, contains no acid, and is made up of plates of iron oxide and nickel oxide

[1] See "The Storage Battery," by A. Treadwell.

[2] *Electrical Review*, May 2nd, 1913.

in an alkaline solution. It is stated that vibration does not affect it, and that it may be short-circuited without harm resulting. It is also claimed that it has nearly double the capacity of a lead battery of the same weight.

The vehicle to which these cells were fitted for the R.A.C. test had the following characteristics :—

The weight of the wagon, which had a canvas top delivery body, was as follows—

Chassis weight with battery	24 cwt.
Body weight (unladen)	5 ,,
Total weight	29 ,,
Distance run	312·3 miles
Electricity supplied to battery for this work..	191,810 watt-hours
Average weight of load ..	about 8 cwt.
Average total weight	1·85 tons

The ton-miles run were, therefore, $312{\cdot}3 \times 1{\cdot}85 = 578$.
So that the watt-hours per ton-mile

$$= \frac{191{,}810}{578} = 332.$$

Now, as was shown on p. 20, 2 watt-hours per ton-mile are equivalent to 1 lb. per ton of tractive effort, and if the tractive resistance at the average speed of motion (11·52 m.p.h.) were, say, 80 lb. per ton, the watt-hours per ton-mile required would be 160. Now 332 were supplied to the battery, showing that the overall efficiency from input of battery to motion shaft was about 50 per cent.

If the motors had an efficiency of 80 per cent., then the efficiency of the battery would be about $\frac{50}{0{\cdot}8}$, or about 60 per cent. It so happens that this is precisely the watt-hour efficiency figure claimed by the Edison Company, and the impartial judgment of the R.A.C. test has,

therefore, borne out the claim made for the battery in this respect. The only work done to the battery during this run of 312 miles was to test the level of the electrolyte in some of the cells on two occasions and to add distilled water to all the cells three times.

If the figure 330 watt-hours per ton-mile be taken as normal, the cost per ton-mile of supplying power obtained at $\frac{1}{2}d.$ per unit would be

$$0{\cdot}330 \times 0{\cdot}5 = 0{\cdot}167d. \text{ per ton-mile,}$$

or about a farthing per car mile.

The best figure obtained for petrol vehicles is about 70 ton-miles per gallon, and taking the gallon to cost, say, 17*d.* the cost per ton-mile would be about 0·25*d.*, showing that the electric vehicle should be able to hold its own so far as power costs are concerned.

Trolley Omnibuses—These vehicles resemble electric tramcars in the manner in which they take in current from an overhead conductor, and motor omnibuses in general structural form and the tiring of the wheels.

The disadvantage that more tractive resistance is needed on roads than on rails is neutralised by the advantage that it is less expensive to provide for the extra cost of the electric power needed than to pay for the cost of putting down rails and maintaining them. These vehicles have not yet been running long enough to enable complete comparisons with the existing electric tramways to be made, but they are reported to cost no more per car mile, and they do not involve the large capital commitment necessary in tramway schemes.

They were first used in this country in Leeds, Bradford and Dundee, but experience on the Continent is somewhat longer. Various means are in use for connecting the electric side of the vehicle to the overhead trolley. It is less easy to arrange than on a tramcar, since the trolley omnibus may have to move, when passing traffic, from one

side of the road to the other. What is needed is some erection on the vehicle high enough to enable the line joining it to the overhead trolley to clear the highest vehicles that are likely to be passed, and some method of paying out the line to provide the necessary ability to move across a wide road. The system presents no special problems of scientific interest, since its electrical side is so similar to that of the electric tramcar which has already been brought to a high state of perfection.

HYDRAULIC TRANSMISSIONS—A number of hydraulic transmission gears have been invented, and some have been put into practical use. They are much alike in principle, although the details are dissimilar.

A pump is driven by the engine, and the oil, which is almost always the fluid pumped, passes from the pump to the motor or motors connected to the road wheels. The pump is usually multi-cylinder, having pistons connected to a crank with a variable throw. The motors which receive the oil are also multi-cylinder, but the pistons have a fixed stroke.

The effect of giving the pump-crank a variable throw is to vary the amount of oil pumped per revolution. But if the work done by the engine per revolution be kept constant, the effect of lowering the volume of oil pumped must be to increase its pressure in the same ratio. The engine yields constant power, and the oil pumped must have the product of its pressure by its volume constant—that is assuming that any loss from friction remains sensibly constant.

The motors, therefore, receive oil in varying quantity, and varying pressure. When the quantity is halved the motor speed will be halved and the pressure will be doubled, so doubling the road wheel torque. Consequently the product of road wheel speed by road wheel torque will remain constant, and the graph of the two

will be a rectangular hyperbola which, as already explained, is the ideal torque curve.

Omitting for the moment, therefore, all consideration of frictional or leakage loss, the hydraulic system of power transmission is one by which the ideal is attainable. In practice, of course, losses do occur through friction and leakage. Leakage must obviously occur when the volume of the flow approaches zero, as the pressure then becomes very great.

It is plain that this system has also the advantage of giving an easy reversing motion, since the shortening of the crank throw can continue until it is zero and then begins to grow in the opposite direction. This is equivalent to a 180 degrees change of phase, and reversal accordingly occurs. Regenerative braking is not usual, as there is no means of storing the power unless large hydraulic or pneumatic reservoirs are carried for the purpose.

Among the best known of these systems are the Hele-Shaw,[1] the Manly,[2] and that jointly devised by H. D. Williams and R. Janney.[3]

The Hele-Shaw pump is of a particularly ingenious design, and its efficiency is stated to be over 70 per cent., measuring efficiency as the ratio of road wheel H.P. to engine B.H.P.

[1] *Engineering*, June 21st, 1912.
[2] Am. Soc. M.E., December, 1911.
[3] *Engineering*, January 31st, 1913.

APPENDIX I

ENERGY STORED IN MOVING VEHICLE

The energy stored is of two kinds :

(1) Translational.

(2) Rotational.

The former is equal to

$$\frac{1}{2}\frac{W}{g}V^2 \text{ ft.-lb.}$$

where W is the weight of the vehicle in lb. ; V the speed in ft. per sec. and $g = 32{\cdot}2$.

The latter is equal to the sum of the rotational energy in all the rotating parts, and in any given rotating part it is equal to

$$\frac{1}{2} I\, \omega^2 \text{ ft.-lb.},$$

where I is the moment of inertia of the rotating part and ω is its angular velocity in radians per second.

Now on a motor vehicle there are a great number of rotating parts, but few of them are heavy enough to affect the total rotational energy of the vehicle. The only parts that need be considered are :

(1) The road wheels.

(2) The fly wheel.

When a car is coasting with the clutch out or in neutral gear, the fly wheel is not being driven by the car, and we have only to consider the road wheels. When, however, the car is starting up from rest we may have to take (2) into account. The former is, however, by far the most important case, and it will be chiefly considered. Now the mass of the road wheels is mainly concentrated in the rim and tire, so that to a first approximation we may

regard the whole weight of the wheel as being concentrated at the circumference. This, of course, exaggerates the effect and gives us an outside figure for the rotational energy stored up in this way. Let w be the weight of one of the road wheels and D its diameter in feet.

Then $I = \frac{w}{g}\left(\frac{D}{2}\right)^2$, assuming the whole of the weight to be concentrated in the rim,

$$\text{and } \frac{D}{2} \times \omega = V.$$

$$\text{Therefore rotational energy} = \frac{1}{2} I \omega^2$$

$$= \frac{1}{2}\frac{w}{g}\frac{D^2}{4}\frac{4V^2}{D^2}$$

$$= \frac{1}{2}\frac{w}{g}V^2 \text{ ft.-lb.}$$

$$\text{But the translational energy} = \frac{1}{2}\frac{W}{g}V^2 \text{ ft.-lb.}$$

Therefore the ratio

$$\frac{\text{rotational energy}}{\text{translational energy}} = \frac{\frac{1}{2}\frac{w}{g}V^2}{\frac{1}{2}\frac{W}{g}V^2} = \frac{w}{W}.$$

Showing that the fractional increase in energy due to the rotating road wheels is equal to the ratio of the weight of the rotating wheels to the weight of the whole vehicle. If, for instance, the weight of the wheels be 5 per cent. of the weight of the whole vehicle, then the total energy is 1·05 times the translational energy of the vehicle. This is a first approximation. It is not strictly true, since the whole mass of the wheels is not concentrated at the circumference, but is within it. But we get in this way an upper limit. It shows that we may roughly regard the wheel weights as counting twice in the energy account —first because they are moving with the vehicle, and secondly because they are rotating with a peripheral speed equal to the speed of the vehicle. So the effective

weight is greater than the dead weight, and if a car be coasting on a level road, the rotational energy will help to keep it going just as much as the translational energy, and we have in consequence to think of the effective weight rather than of the dead weight.

The above method of finding the ratio of effective and dead weight is only a rough and ready one. We may calculate the exact value from the drawings—a somewhat tedious process—or we may get it experimentally. We might, for instance, take the accelerometer reading when coasting at any convenient speed along a level road and compare this with the reading when coasting down a slope (having the same road surface), such that the velocity was just maintained at the same figure, then the ratio of these two readings would give the required result. This is manifestly the case, since on the level we have to think of the effective weight, whilst down the slope, where there is no change of speed, there can be no increase or decrease of rotational energy, and we need only think of the dead weight (which of course is changing its potential energy). We will again refer to this method of measurement, which, however, is not very easily carried out owing to unavoidable differences of road surface on hills and on level roads; but in the meantime there is another very effective and simple way of measuring the rotational energy of the wheels experimentally. It is a method of measuring the moment of inertia.

If the vehicle be raised so that the road wheel it is proposed to test is freed from contact with the ground and the wheel be slowly turned and then left to settle, it will be found that it always comes to rest with one point on the wheel downwards, due probably to the weight of the tire valve, etc., or to some other inequality. A small weight is now attached to the opposite point of the wheel, so that this excess is just balanced. The wheel will now come to rest in any position. Let the small weight added be w_1 and its radius (in feet) from the centre of rotation r_1. Now add another weight, w_2, at any convenient radius r_2, and allow the wheel

to oscillate to and fro as it comes to rest with this weight at the bottom. Take the time of a complete oscillation and call it t seconds, then it can be shown that

$$t = 2\pi \sqrt{\left(\frac{w_1 r_1^2 + w_2 r_2^2}{g} + \mathrm{I}\right) \frac{1}{w_2 r_2}}$$

or

$$\frac{w_1 r_1^2 + w_2 r_2^2}{g} + \mathrm{I} = w_2 r_2 \frac{t^2}{4\pi^2},$$

therefore

$$\mathrm{I} = w_2 r_2 \frac{t^2}{4\pi^2} - \frac{w_1 r_1^2 + w_2 r_2^2}{g}.$$

In a case tested by the author, the following data were obtained :—

$$w_1 = 0$$
$$w_2 = 7 \text{ lb.}$$
$$r_2 = 10 \text{ inches} = 0{\cdot}833 \text{ ft.}$$
$$t = 3{\cdot}05 \text{ secs.}$$

so that

$$\mathrm{I} = 7 \times 0{\cdot}833 \times \frac{(3{\cdot}05)^2}{4\pi^2} - \frac{7 \times (0{\cdot}833)^2}{32{\cdot}2}$$
$$= 1{\cdot}22.$$

This would be equal to a weight of 22 lb. concentrated at the rim of the wheel (810 mm. diam.), or 88 lb. for four similar wheels—being 2½ per cent. on a car weight of about 1½ tons, and giving the ratio of effective to dead weight when coasting declutched of 1·025.

For this same car an investigation showed that the corresponding figure for the inner part of the clutch was only one-fifteenth of that of one road wheel even when allowance was made for its more rapid rotation than the road wheel (top gear figure)—showing it to be a negligible quantity. A calculation of the fly-wheel showed that on top gear the momentum stored was 1½ per cent. of that of the car. On bottom gear it would be about 5 per cent. of that of the car—showing a total rotational momentum equal to about 7½ per cent. of the translational momentum when on bottom gear. The ratio of the two *energies* is, however, in the neighbourhood of 20 per cent. when on bottom gear.

The momentum figure is, however, more important than the energy figure, since the rate of change of momentum per ton is a measure of the tractive effort in lb. per ton.

Problem—Let a vehicle of weight W be coasting down a slope of 1 in s, and let the accelerometer reading be R in lb. per ton. Let ratio of rotational to translational momentum be m. What is the true value of the tractive resistance in pounds per ton? Let this true value be T. Then, accelerating force down the slope

$$= 2{,}240\,\frac{W}{s} - WT \text{ lb.}$$

Now effective mass $= \dfrac{W}{g}(1+m)$ tons.

$$\text{So that acceleration} = \frac{\dfrac{2{,}240W}{s} - WT}{\dfrac{2{,}240W}{g}(1+m)} = \frac{(2{,}240 - T\,.\,s)}{2{,}240s\,(1+m)}\cdot g$$

The accelerometer will subtract this acceleration from that corresponding to the slope, and the net acceleration read will be

$$\frac{g}{s} - \frac{(2{,}240 - T\,.\,s)}{2{,}240s\,(1+m)}\cdot g$$

$$= \frac{g}{s}\left\{1 - \frac{2{,}240 - T\,.\,s}{2{,}240\,(1+m)}\right\}$$

$$= \frac{g}{s}\left\{\frac{2{,}240 + 2{,}240m - 2{,}240 + Ts}{2{,}240\,(1+m)}\right\}$$

And since the reading R is in lb. per ton we have

$$R = \frac{2{,}240}{g}\cdot\frac{g}{s}\left\{\frac{2{,}240m + Ts}{2{,}240\,(1+m)}\right\} = \frac{2{,}240m + Ts}{s\,(1+m)}$$

therefore

$$sR\,(1+m) = 2{,}240m + Ts$$

or

$$Ts = -\,2{,}240m + sR\,(1+m)$$

$$T = R\,(1+m) - \frac{2{,}240m}{s}$$

$$= R + m\left(R - \frac{2{,}240}{s}\right)$$

showing that if the slope be such that $\frac{1}{s} = \frac{R}{2,240}$ then $T = R$, and there is no correction whatsoever to apply. If we put m at 5 per cent., then we have

$$T = R\left\{1 + m\left(1 - \frac{2,240}{sR}\right)\right\}$$

$$= R\left\{1 + 0{\cdot}05\left(1 - \frac{2,240}{sR}\right)\right\}$$

and the fractional error in the reading will be

$$0{\cdot}05\left(1 - \frac{2,240}{sR}\right)$$

if $R = 150$ lb./ton, and $S = 1$ in 20, then the fractional error is

$$0{\cdot}05\left(1 - \frac{2,240}{20 \times 150}\right) = 0{\cdot}05\left(1 - \frac{2,240}{3,000}\right)$$

$$= 0{\cdot}013$$

or about $1\frac{1}{4}$ per cent., which is below the errors of observation, and is, therefore, negligible.

Were it a matter of *ascending* this grade the error would *exceed* 5 per cent., and would be

$$0{\cdot}05\left(1 + \frac{2,240}{3,000}\right) = 0{\cdot}088$$

or about $8\frac{3}{4}$ per cent.

So uphill readings are liable to error—but as they are practically impossible to take the point is not one of much practical consequence. If we take two readings, one up and one down, the same slope and at the same speed, and if we call these readings R_1 and R_2, we have

$$R_1\left\{1 + m\left(1 + \frac{2,240}{sR_1}\right)\right\} = R_2\left\{1 + m\left(1 - \frac{2,240}{sR_2}\right)\right\}$$

and

$$\frac{R_1}{R_2} = 1 + m\left(1 - \frac{2,240}{sR_2}\right) - m\left(1 + \frac{2,240}{sR_1}\right) \text{ approximately,}$$

so that

$$\frac{R_1}{R_2} = 1 - m\frac{2,240}{sR_2} - m\frac{2,240}{sR_1}$$

or

$$= 1 - \frac{2{,}240m}{s}\left(\frac{1}{R_2} + \frac{1}{R_1}\right)$$

$$= 1 - \frac{2{,}240m}{s}\left(\frac{R_1 + R_2}{R_1 R_2}\right)$$

$$\therefore \frac{2{,}240m}{s}\left(\frac{R_1 + R_2}{R_1 R_2}\right) = 1 - \frac{R_1}{R_2} = \frac{R_2 - R_1}{R_2}$$

$$\therefore \qquad m = \frac{s}{2{,}240}\left\{\frac{R_2 - R_1}{R_1 + R_2} \cdot R_1\right\}.$$

This it will be seen affords a method of measuring m experimentally.

APPENDIX II

ROYAL AUTOMOBILE CLUB — EXPERT AND TECHNICAL COMMITTEE

(See *R.A.C. Journal* for September 20th, 1912 and November 15th, 1912.)

REPORT ON THE BRAKE HORSE-POWER TESTS AT BROOKLANDS ON JULY 19th, 1912.

METHOD OF TESTS.

1. The method of carrying out the tests was that described in the *R.A.C. Journal*[1] for July 26th, 1912.

Twenty-seven cars were tested and the results are set out in Table I. It will be seen that a number of old cars

[1] The following is an extract from the *R.A.C. Journal* of July 26th, 1912:—

"Starting from the paddock, each car, furnished with a Wimperis accelerometer, entered the oval of the track at the 'fork,' and was driven at its maximum speed over the measured half-mile alongside the railway. About 10 yards beyond the measured stretch, with the car still going at its maximum speed, the clutch was disengaged, thereby causing the car to slow down, and the rate at which deceleration took place supplied the important factor of the test. The Wimperis instrument is graduated to show the deceleration or retardation of the vehicle in pounds per ton, and, if the figure indicated by the accelerometer pointer be employed in the formula

$$\frac{\text{Velocity (m.p.h.)} \times \text{Retardation (lb. per ton)} \times \text{Total weight of vehicle}}{375}$$

the simplification of the latter will give the horse-power developed by the engine when propelling the car at its highest speed. The cars used for the experiments differed widely in horse-power and weight, and the results when known should prove distinctly educational.

"The ease with which each car was tested may be gathered from the fact that all of them were dealt with in about four hours. No delicate adjustments were necessary, beyond the levelling up of the accelerometer on the vehicle."

were included, and for the first time it has been possible to ascertain the relative H.P. of a number of new and old cars of various sizes and makes. The B.H.P. figures are obtained from the formula

$$\text{B.H.P.} = \text{R W V} \div 375$$

where R is the resistance to motion measured at the clutch in pounds per ton, W is the car weight in tons, and V is the speed in m.p.h.

Gear Ratios and Wheel Diameters.

2. On Dr. Dugald Clerk's suggestion, the Club has endeavoured to obtain from the owners of the cars which ran on July 19th particulars of gear ratios on top gear, diameters of rear road wheels, and information as to whether the top gear was a direct drive or not. Table II gives the additional data so collected. It has, of course, not been verified by the Club. The purpose for which these additional particulars were desired was to get figures for the engine r.p.m. at top speed, the piston speed at that point, and the brake mean effective pressure (ηP).

Comparison with R.A.C. Rating and Effect of Age.

3. A point of interest in the tests is that it enables a comparison to be made of B.H.P. with R.A.C. rating, not only for new cars but for old ones. The ratio of B.H.P. to R.A.C. rating is given for each car in Table III. It will be seen that the B.H.P. is greater than the R.A.C. rating in the case of 1911 and 1912 cars in the ratio (mean) of 1·46 to 1. This ratio sinks to 1·18 in the case of cars built in 1909 and 1910; to 0·76 for 1907 and 1908 cars; and to 0·55 for cars six years old and over. The loss in relative power in passing from the 1909–10 group to the 1907–8 cars is marked, and indicates an advance in motor car design in the intervening period. It will be seen that cars four or five years old give on the average half the B.H.P. of new cars of the same R.A.C. rating.

GENERAL RESULTS.

TABLE I.

Car No.	Maker's H.P.	Body Seats.	Date.	Bore & Stroke. mm.	No. of Cylinders.	R.A.C. Rating.	Displacement C.C.	Weight as Run. Tons.	Maximum Speed. m.p.h.	R. lb. per ton.	B.H.P.
1	8	4	1905	127 × 127	1	9·9	1,583	1·04	24·0	90	6
2	15	4	1908	98 × 114	4	23·8	3,440	1·32	36·44	135	17½
3	12–16	4	1912	79 × 121	4	15·5	2,352	1·36	42·45	155	24
4	14–20	4	1912	80 × 130	4	15·9	2,610	1·44	36·73	135	19
5	12	2	1912	70 × 120	4	12·1	1,843	1·30	38·14	150	20
6	24–30	Closed	1911	100 × 130	4	24·8	4,082	1·88	40·72	120	24½
7	15	4	1911	70 × 170	4	12·1	2,611	1·20	44·33	185	26
8	12–16	4	1910	79 × 114	4	15·5	2,235	1·25	40·36	118	16
9	10–12	4	1909	102 × 110	2	12·9	1,797	1·18	28·75	92	8½
10	10–12	2	1912	75 × 110	4	13·96	1,940	1·08	37·19	175	18½
11	16–20	4	1907	95 × 135	4	22·4	3,828	1·46	37·97	130	19
12	20–28	4	1911	102 × 130	4	25·8	4,248	1·67	44·78	165	33
14	15	4	1912–13	89 × 127	4	19·6	3,170	1·62	38·96	145	24½
15	18–30	4	1911	100 × 130	4	24·8	4,082	1·91	39·47	110	22
16	26	4	1912	102 × 178	4	25·8	5,818	1·98	49·72	165	43½
17	20	4	1910	102 × 140	4	25·8	4,575	1·45	56·25	220	48
18	12–16	2	1907	82 × 120	4	16·7	2,535	1·00	41·10	105	11½
19	15	4	1910	85 × 110	4	17·9	2,497	1·36	34·09	125	15½
20	18	4	1912	90 × 130	4	20·1	3,307	1·67	44·78	175	33½
21	12	4	1912	75 × 130	4	13·96	2,298	1·21	40·00	140	18
22	12	2	1903	80 × 120	4	15·9	2,409	1·05	24·52	120	8
23	Not given	2	1909	106 × 170	4	27·8	6,001	1·71	46·88	130	27½
24	12–16	2	1912	80 × 150	4	15·9	3,012	1·20	51·43	175	29
25	12	4	1912	75 × 130	4	13·96	2,298	1·26	43·48	195	28½
30	15	4	1910	80 × 120	4	15·9	2,409	1·38	37·66	160	22
32	15·9	2	1912	80 × 120	4	15·9	2,409	1·35	43·27	115	18
33	15	4	1905	90 × 120	4	20·1	3,052	1·37	30·3	100	11

Effect of Stroke-bore Ratio.

4. Some engineers have considered that the greater H.P. got from modern cars is due entirely to increase in the length of stroke, and that the H.P. might be expected to increase in step with increase in stroke-bore ratio. Table IV has therefore been constructed to show the relationship between these two ratios. Careful examination of the figures shows that the relationship, if any, is distant.

Engine and Piston Speeds and ηP.

5. In Tables V, VI and VII are given the engine and piston speeds and the values of ηP for new (1911–12), middle-aged (1908-10), and old (1903–7) cars. The engine speed (mean) is 1,702 for the new, 1,440 for the middle-aged, and 972 for the old. There seems, therefore, to have been some little increase in engine speed in the last three or four years, and a material advance between the old and middle-aged. The low engine speed of the old cars is, however, due in part to the fear of damage to engine or car if the throttle were opened to its full extent. The mean piston speed for new cars is 1,478 feet per minute, compared with 1,160 for the middle-aged, and 817 for the old. The figures for ηP are striking. The new cars give a value of ηP almost identical with that assumed in the R.A.C. rating, viz., 67·2. Middle-aged and old cars have about the same mean pressures, viz., 51 and 54, suggesting that the loss of pressure due to piston and valve leakage and to lower mechanical efficiency gets to its maximum when the car is still only middle-aged, and that the further drop in H.P. is due to the lowness of piston speed of the very old cars.

Air Resistance.

6. Professor Callendar suggested that advantage might be taken of the data obtained to deduce comparative air resistance figures for the cars. This was not contemplated at the time the experiments were made, or care would

have been taken to make a description of the kind of body, of the position of the wind-screen, and to obtain a figure for R when the car was approaching rest. If, however, the value of R at very low speeds be put at 50 lb. per ton for all cars (this is of the nature of a guess, though the figure may fairly confidently be expected to lie between 40 and 60), it is possible to tabulate the results as shown in Table VIII. Professor Callendar's suggestion was to assume that—

Total resistance in pounds $= RW = aW + cV^2$ where a is some constant, and c a figure proportional to the effective wind area of the car. Dividing through by W

$$R = a + \frac{c}{W}V^2$$

$$\text{or } (R - a)\,W \div V^2 = c.$$

In Table VIII the last column contains the value of the above expression (taking $a = 50$) and very roughly gives the wind resistance areas. The wind resistance of a car is greater for a four-seater body than for a two-seater, as the back seats of the former account for a substantial part of the resistance. Two-seater cars have therefore been omitted from the table. It will be seen that the average value of c is $8{\cdot}28 \div 100$, or $0{\cdot}0828$. Now the ordinary wind pressure formula is

$$\text{Pressure in lb./sq. ft.} = 0{\cdot}003\ V^2,$$

so that the mean effective area of surface exposed to the wind would be 27½ square feet. This calculation omits, however, two important considerations :—

(*a*) The pressure formula assumes normal surfaces, whereas most of those in the car are sloping, and it omits skin resistance and pocketing.

(*b*) Part of the V^2 term is undoubtedly due, not to air resistance, but to churning up oil in the gear-box.

Following the lines of Professor Callendar's suggestion, the average formula for resistance for the cars in Table VIII would be

$$R = 50 + \frac{8 \cdot 3}{W}\left(\frac{V}{10}\right)^2$$

and

$$\text{B.H.P.} = \frac{RVW}{375} = \frac{VW}{7 \cdot 5} + \frac{V^3}{4{,}500}$$

BRAKE HORSE-POWER TESTS.

TABLE II.

No. of Car.	No. of revs. of engine to 1 of road wheels, with Top Speed in.	Diam. of Rear Tires. mm.	Top Speed Direct Drive.	Remarks.
1	4·0	760	Yes	Believes 4 to 1 is correct.
2	9 to 5	810	„	—
3	4·37	810	„	—
4	—	—	—	No reply.
5	4·38	760	Yes	—
6	3·78	920	„	—
7	—	810	„	Does not give revolutions.
8	4·4	810	„	—
9	4·12	760	—	No reply.
10	4·285	760	Yes	—
11	2·88	880	„	—
12	3·3	820	„	—
14	4·0	820	„	—
15	3·37	880	„	—
16	3·25	880	„	—
17	—	—	—	No reply.
18	3·0	760	—	—
19	4·0	815	Yes	—
20	4·0	820	„	—
21	4·7	810	„	—
22	3·0	810	No	4 speeds (indirect).
23	2·5	880	Yes	—
24	3·4	815	„	—
25	4·7	810	—	—
30	—	—	—	No reply.
32	—	—	—	„
33	—	—	—	„

NOTE.—The Gear Ratio for Car No. 2 is obviously incorrectly given, and it is omitted in the calculations.

TABLE III—Giving ratio of B.H.P. to R.A.C. Rating; Dividing Cars in groups according to age.

Car No.	B.H.P./R.A.C. Rating.			
	1912-1911	1910-1909	1908-1907	1906-1903
1	—	—	—	0·61
2	—	—	0·74	—
3	1·55	—	—	—
4	1·19	—	—	—
5	1·65	—	—	—
6	0·98	—	—	—
7	2·15	—	—	—
8	—	1·03	—	—
9	—	0·66	—	—
10	1·32	—	—	—
11	—	—	0·85	—
12	1·28	—	—	—
14	1·25	—	—	—
15	0·89	—	—	—
16	1·68	—	—	—
17	—	1·86	—	—
18	—	—	0·69	—
19	—	0·86	—	—
20	1·67	—	—	—
21	1·29	—	—	—
22	—	—	—	0·50
23	—	0·99	—	—
24	1·82	—	—	—
25	2·04	—	—	—
30	—	1·38	—	—
32	1·13	—	—	—
33	—	—	—	0·55
Average ...	1·46	1·13	0·76	0·55

ACCURACY OF MEASUREMENTS.

7. When these tests were decided upon, it was expected to get readings to within 5 per cent., and it is interesting, in conclusion, to examine into the probable accuracy achieved. The figures for R may be taken as $\pm$ 5 lb. per

TABLE IV—Comparison in case of 15 New Cars (1911 and 1912) of the B.H.P./R.A.C. rating ratio with stroke-bore ratio.

Car No.	B.H.P.	R.A.C. Rating.	B.H.P./R.A.C.	Stroke-bore Ratio, l/d.
3	24	15·5	1·55	1·53
4	19	15·9	1·19	1·62
5	20	12·1	1·65	1·72
6	24½	24·8	0·98	1·30
7	26	12·1	2·15	2·43
10	18½	13·96	1·32	1·47
12	33	25·8	1·28	1·27
14	24½	19·6	1·25	1·43
15	22	24·8	0·89	1·30
16	43½	25·8	1·68	1·75
20	33½	20·1	1·67	1·44
21	18	13·96	1·29	1·73
24	29	15·9	1·82	1·88
25	28½	13·96	2·04	1·73
32	18	15·9	1·13	1·50
Average ...	—	—	1·46	1·60

ton, or, on the average, some 3 per cent. above or below the correct figure. (When tests are made on a downward slope the accuracy is very much greater, as there is more time to make the observations, and the complicating effect of rotational momentum is almost wholly absent.) The momentum stored in the rotating parts of the car tends to keep up the speed of the car when coasting on a level road, and this effect may make the reading of R some 2 or 3 per cent. low. The fact that the "straight" at Brooklands is approached from a downward slope tends to make the speed a little higher than it would be if due to the engine alone, and this tends to make the readings high, thus more or less balancing the effect of rotational momentum. The above considerations apply to the four-seater cars; in the case of the two-seaters the only place for the accelerometer was on the floor-boards, and, as these were not, in general, screwed or

fastened down, there was uncertainty as to whether they may not have shifted. Great care was taken to minimise the chances of error on this account, but all observers can testify to the difficulty of so doing. For this reason the measurements made on the two-seater cars were not so accurate as those made on the four-seater cars, but it is impossible to say precisely to what extent.

TABLE V—Engine r.p.m.; Piston Speed and ηP for 11 New Cars (1911 and 1912).

Car No.	Engine r.p.m.	Piston Speed, feet per min.	ηP.	Stroke-bore Ratio l/d.
3	1,950	1,550	68	1·53
5	1,870	1,470	75	1·72
6	1,430	1,220	55	1·30
10	1,790	1,290	69	1·47
12	1,540	1,310	65	1·27
14	1,620	1,350	62	1·43
15	1,290	1,100	54	1·30
16	1,560	1,820	62	1·75
20	1,860	1,590	71	1·44
21	1,980	1,690	51	1·73
24	1,830	1,810	68	1·88
Average ...	1,702	1,473	64	1·53

TABLE VI—Engine r.p.m.: Piston Speed and ηP for 4 Middle-aged Cars (1908—1910 inclusive).

Car No.	Engine r.p.m.	Piston Speed, feet per min.	ηP.	Stroke-bore Ratio l/d.
8	1,870	1,400	50	1·44
9	1,330	960	46	1·08
19	1,430	1,030	56	1·29
23	1,130	1,270	53	1·60
Average ...	1,440	1,165	51	1·35

TABLE VII—Engine r.p.m.; Piston Speed and ηP for 3 Old Cars (1903—1907 inclusive).

Car No.	Engine r.p.m.	Piston Speed, feet per min.	ηP	Stroke-bore Ratio l/d.
1	1,080	900	45	1·00
11	1,060	940	61	1·42
22	775	610	56	1·50
Average ...	972	817	54	1·31

TABLE VIII—Estimation of Air Resistance Constants for all four-seater Open Cars with open bodies (hoods down, some screens up, some not).

Car No.	W Weight tons.	R Resistance.	V Max. Speed m.p.h.	$(R-50) W \div \left(\frac{V}{10}\right)^2$
1	1·04	90	24·0	7·2
2	1·32	135	36·44	8·4
3	1·36	155	42·45	7·9
4	1·44	135	36·73	9·0
7	1·20	185	44·33	8·2
8	1·25	118	40·36	5·2
9	1·18	92	28·75	6·0
11	1·46	130	37·97	8·1
12	1·67	165	44·78	9·6
14	1·62	145	38·96	10·1
15	1·91	110	39·47	7·4
16	1·98	165	49·72	9·2
17	1·45	220	56·25	7·8
19	1·36	125	34·09	8·8
20	1·67	175	44·78	10·4
21	1·21	140	40·00	6·8
24	1·71	175	51·43	8·1
25	1·26	195	43·48	9·6
30	1·38	160	37·66	10·7
33	1·37	100	30·3	7·4
Average ...	—	—	—	8·28

APPENDIX III

ROAD TEST REPORT FORM

Report on Automobile Road Test

Make and Description of Car—

Engine No. ....
Engine (Make) ....
No. of Cylinders ....
Bore......... Stroke.........
Capacity.............. c.c.s.
Speed (normal)...... r.p.m.
H.P. (R.A.C. rating).........
Carburettor ....
Ignition System ....
Lubrication ,, ...
Water Circulation ....
Velocity ratio of engine to road wheels on each speed ......
....
Velocity ratio of bevel, or worm drive to differential ...
....
Chassis No. ....
Type of Body ....
Weight of Chassis............
Weight of Body ....
Capacity ....
Total Weight (unloaded)...
....

Total Weight (loaded) during Road Test ......
Wheel Base ....
Type of Lamps ....
,, ,, Speedometer ......
,, ,, Clutch ....
Finish (Brass or Nickel)......
Gauge (cre. to cre. of Wheels ......
Clearance under lowest part of Car (say where) ....
Nature of drive...............
if chain, give make ......
type ......
length......
Front Tires ....
Rear Tires ....
Make and type of Road Wheels ....
Make and type of Wind Screen ....
Type of Hood or Canopy ....
Colour of Paint...............
,, ,, Trimming.........

Road Test. (*Date*.................)—

State of Roads ....
Load carried on Trial ......
Distance Run .... miles
Tires Used.....................
Measured Speeds ....
....
Fuel Used ....
Fuel Consumption......mls. on.........gall.
mls. per gall.=.........
"Gross Ton Miles" per gall. =......
Temp. in water at top of radiator after steady run at full power =......
Temp. in Shade =...........
Tests on Brakes (pounds per ton) ......
Notes on working of Clutch ......
Notes on working of Change of Gear ......

Notes on working of Springs ......
Diam. of circle inside which Car can turn in either direction without reversing ......
Average Tractive Resistance (at clutch) ......
"Gross Ton Miles" per gallon of fuel reduced to basis of 70 lb. per ton T.R. ......
Maximum Grades Climbable on each Gear, determined by accelerometer:—
....
....

Remarks.

....
....
....

........................*Observer.*

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